Understanding the field performance of domestic heat pumps

An analysis of recent residential heat pump field trials and training needs

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A thesis submitted in partial fulfilment of the requirements of University College London for the degree of Doctor of Philosophy, February 2014.

‘ I, Colin Patrick Gleeson, confirm that the work presented in this thesis is my own. Where information has been derived from other sources, I confirm that this has been indicated in the thesis.'

Signed

Date

Abstract

This thesis explores heat pump performance. Renewable technology, based on ambient conditions, is at a distinct thermodynamic disadvantage when compared with such technologies as gas condensing boilers since the temperature gradients in which they work are so much smaller. This disadvantage makes renewable technologies, and specifically heat pumps, sensitive to design and installation practice.

A mixed methods approach of quantitative and qualitative investigation is applied, principally through the analysis of heat pump field trial performance; a meta-analysis of eight European field trials of over 600 heat pump installations in terms of historical and contemporary system boundaries, and a taxonomical analysis of the UK Energy Saving Trust field trial. The trials are placed in context through the analysis of UK central heating practice, UK and EU policy, thermodynamics, manufacturers' test regimes and a pilot field trial.

From this analysis it is apparent that a wide range of performance is exhibited by residential heat pump installations. This potential to underperform, or 'sensitivity to context', is explored through its plausible link to vocational education and training (VET). The process of re-aligning EU VET for heat pumps is underway, driven in the UK by the Microgeneration Scheme's design literature and training requirements. However, doubts remain as to the abilities of current UK contractors to synthesise the technical design requirements given the relatively low educational demands made on residential heating occupations when compared with EUCERT heat pump requirements, more closely aligned with the Continental concept of *savoir-faire*, 'know-how' or *berufliche Handlungsfähigkeit*, a multidimensional 'occupational capacity'.

Governments, manufacturers and contractors wishing to embrace renewables, especially heat pumps, must recognise that 'context' is multi-dimensional, a complex amalgam of national historical development, engineering thermodynamics, product evolution and installer knowledge, skills and competence. The thesis attempts to map these themes in response to field trial evidence.

Acknowledgement:

Sincere thanks to Steve Fuller of EDF-Energy for generously releasing the EST trial raw data files, to my supervisor Professor Robert Lowe from whom I learned so much, to Dr Ian Ridley and to Barratt Developments.
I also extend my thanks to my examiners, Dr Roger Nordman of SP Technical, Sweden and Dr Dejan Mumovic of UCL, whose suggested corrections have undoubtedly strengthened the thesis.

To Maggie Stack & Cis Manton Gleeson

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Introduction

This thesis proposes that the successful introduction of any renewable technology that relies on the relatively low heat fluxes associated with ambient temperatures cannot depend solely on government directives supported by financial sweeteners. Such technologies demand a far more exacting attention to design, installation and operation than that currently applied to systems based on fossil fuels and their success is dependent on installer vocational education and training along with the promotion by manufacturers of transparent performance data and "user-friendly" appliance control systems.

In order to reach this conclusion, the research journey commences in Chapter 1 with an overview of the UK domestic heating market, the dominance of the gas boiler and, in particular, the combination boiler or the "combi". The combi is a mature product that has been developed to provide simplicity of installation, automatic modulation of heat output matched to system demand, built-in user controls and fault diagnosis. It is into this scenario that the heat pump, perhaps the most complex of domestic heating appliances, is being promoted by EU and UK Government policy decisions. The drive to increase the uptake of heat pumps is premised on their potential to contribute increasing levels of renewable heat as the carbon intensity of grid electricity reduces over the coming years.

Chapter 2 provides a description of the research methodology based on a mixed methods approach. Heat pump technology is contextualised in the early chapters in order to position the data analysis from continental European and UK trials. The initial exploratory data analysis of Energy Saving Trust (EST) field trial raw data indicates that a more complete analysis requires the combination of both quantitative and qualitative methods. The EST data provides population, group and individual system outputs that support a sociological interpretation of trial results in terms of vocational education and training.

Chapter 3 reviews the fundamental thermodynamics of the reverse heat engine, the heat pump, identifying the potential to raise low temperature energy to a temperature sufficiently high to provide both space heating and domestic hot water at efficiencies that reduce carbon dioxide emissions in comparison with conventional heating systems. It is necessary, therefore, to understand the underlying principles of the heat pump if this efficiency is to be realised since the driving forces for energy transfer in heat pumps, the temperature gradients, are radically lower than those for boilers. However, much of what constitutes classical thermodynamic studies is of limited use to the practitioner in that it deals with steady state conditions and whilst concepts such as the Clausius inequality, reversibility and entropy offer a vehicle for analysis of potential efficiencies, in the absence of an "entropy meter", alternative approaches are required. Perhaps the most useful expression of heat pump performance is "Carnot efficiency" where the maximum practical efficiency is related to the temperature of the source and sink reservoirs, that is, the ground loop or ambient air temperature and the emitter temperature. All additional efficiency losses can be assigned to three causes: the entropy generated by any practical thermo-mechanical device, poor system design or poor system control. The chapter continues with an analysis of pressure-enthalpy diagrams and their relevance to heat pump performance, the design of the vapour compression cycle, the heat pump components and the role of high-efficiency components such as variable speed compressors and electronic thermostatic valves and their relationship to the EST trial population. Finally, the chapter reviews the need for dynamic modelling to express seasonal rather than cycle efficiency.

Chapter 4 reviews laboratory-based heat pump testing in order to assess potential performance in the field. Heat pump operating efficiency or 'coefficient of performance' (COP) is provided by manufacturers through EN 14511 laboratory testing designed to simulate space heating performance based on full load conditions at constant temperatures and thus does not reflect annual operation. Several methods for testing space heating and domestic hot water COP are described, including a pilot study carried out by the author at the Barrett Green House, BRE, UK. The process of monitoring is itself fraught with potential hazards that require the monitoring designer to understand the mechanics of the heat pump, the monitoring equipment and the data analysis methods required to assess performance. These test results may be utilised to provide an annual or 'seasonal coefficient of performance' (SCOP), that is, a modelled 'seasonal performance factor' (SPF) for a particular heat pump in a particular climate zone. Such an approach is supported by EN 15316 although the results are derived from full load operation centred on EN 14511 COP test temperatures and 'binned' weather data for the installation location. Outdoor temperatures are collated based on the bin method, a histogram approach, and divided into broader temperature bands centred EN 14511 test temperatures. Manufacturers' COP test results then provide a weighted assessment of seasonal COP for space heating. Further complicating this 'bin method' approach is the issue of domestic hot water, its comparative load size in comparison to space heating, whether the heat transfer process is modelled as steady state, heating from cold or re-heating after tapping, and the appropriate storage temperature. Manufacturers generally only provide domestic hot water data for "compact" units such as for Passivhaus application, so for most installations the hot water must be modelled using data from steady state space heating at fixed flow and return temperatures. Importantly, it must also be recognised, that the bin method model assumes perfect installation and perfect controls and so the procedure cannot be said to reflect real world conditions.

The assessment of heat pump SPF, it would appear, is therefore dependent on field trial studies where heat pumps are tested against dynamic loads dependent on dynamic weather conditions and where heat pump central heating systems reflect the current state of design and installation. Such trials depend on the selection of an appropriate sample chosen to reflect the key objectives of policy, that is, the replacement of domestic boilers producing space heating and domestic hot water with monovalent heat pumps or, where manufacturers provide bivalent units, the separate role of resistance or any alternative fuel backup. The analysis of trial outputs requires a monitoring protocol that recognises the complexity of installing probes and sensors for different manufacturers' products, a process that requires a detailed engineering understanding of the component parts and internal controls, bivalency and, most importantly, a recognised 'system boundary' at which SPF is established.

Chapter 5 outlines the search within this thesis for an internationally recognised system boundary and thus an SPF that can support international comparisons of heat pump studies. This search has resulted in a peer-reviewed paper based on an analysis of eight European heat pump field trials comprising air, ground and water source systems supplying water-based heating. Thirteen different system boundary-naming conventions are identified for heat pump trials lasting for a minimum of one year and consisting of over 600 installations. The review of monitoring methodologies indicates that seven of these boundaries are unique and that trial results may be quoted in any one of the seven, a situation not unlike the proverbial Tower of Babel for specifiers and clients.

Analysis of the published average efficiency for each trial shows that no simple mathematical approach can be identified that would support unproblematic quantitative comparison of performance estimates based on these seven different boundary results and provide a single expression of SPF.

A heat pump-based wet central heating system comprises the source fan or ground loop circulator, the electrical components of the heat pump itself, any integrated backup electrical resistance heater and the circulation pump. A single boundary efficiency metric is fraught with conflict between manufacturers who wish to be able to publish the highest measure of efficiency for their product, and the end user who is interested in how much energy they have to will pay for to operate the system as a whole. The outcomes from the EU-funded SEPEMO trials in identifying a pan-European boundary naming convention go some way towards simplifying the existing situation. But the work demonstrates that for complete transparency of whole system operation, measurements at four boundary conditions are required. However, measurement of efficiency is not an end in itself; it provides the ability to interrogate system operation and the role of individual components in the search for optimised operation and provides feedback for future design and installation.

Chapter 6 provides an analysis of the first year of the Energy Saving Trust's UK heat pump field trial based on the full data set made available by Trial sponsor EDF UK. When taken as a whole, unsurprisingly, the data confirm the reported poor performance of these systems. In order to provide a deeper understanding of the underlying reasons, a taxonomy of system design and monitoring protocol is developed for both ground and air source types based on the Carnot expression of efficiency.

Whilst trial seasonal efficiencies provide some evidence of the veracity of this taxonomical approach, the analysis of system morphologies indicates fundamental flaws in the design of the trial if the outcomes were to provide definitive proof of the heat pump as an alternative to the central heating boiler, that is, a system providing both space heating and domestic hot water. For these conditions, of the 51ground source heat pumps analysed only 7, and of the 24 air source heat pumps only 1, provide boundary monitoring that identifies the heat pump efficiency alone. All other systems are hybrids with combinations of space heating only, space heating with backup, space heating and hot water including circulation pump or hot water measurements based on draw-off rather than heat into the hot water cylinder. This variety of system morphologies is presented as a single "system efficiency" (SEFF) metric. The trial, in effect, directly meets the 'boiler replacement with heat pump central heating' criterion in only 8 out of 75 cases and cannot therefore reasonably claim to measure the spread of seasonal efficiencies of heat pumps as an alternative to the UK default, the gas boiler.

The trial data can however be utilised to identify critical system design criteria such as heat pump sizing, ground loop length, use of buffer vessels, impact of rapid cycling or 'hunting'. The limitations of the trial monitoring protocol provide the main obstacle for such an analysis since there is no consistent underlying logic in the collection of data where, for example when considering all installations (both space heating only and space heating with domestic hot water), in only 10 of the 51ground source and just 2 of the 24 air source heat pumps is the efficiency of the heat pump alone monitored. The use by the trial designers of the single efficiency metric SEFF obscures rather than exposes the role of the system components. The analysis by taxonomy of monitoring protocol does, however, provide sufficient evidence to indicate serious failings in system design attributable to such fundamentals as matching heat pump power output to design heat loss, the sizing of ground loop length, inappropriate flow temperatures and inadequate controls.

Although the analysis by taxonomy is unique, the general failure in design and installation has been noted in previous reports by both the EST and DECC and has lead to training requirements for installer registration through the Microgeneration Scheme (MCS).

Chapter 7 explores the standards, documentation and web support provided by the MCS in response to the criticisms of the design and installation quality of EST trial heat pumps. MCS demand minimum standards of quality assurance and training as a prerequisite for company registration and to support the training aspects have developed a significant body of online technical support including spreadsheets, software and design guides. For those versed in the nuances of room-by-room heat loss calculations, instantaneous power and annual energy, pipe and pump sizing based on a minimum Reynold's Number - all in conjunction with manufacturers' appliance technical data, such a rich literature is sure to raise existing standards. Example calculations identify the need for a high level of technical knowledge, a "thermal literacy" that is acknowledged to be lacking in many who currently design and install, both the formally qualified and the significant fraction of unqualified that make up the workforce.

Minimum entry requirements for heat pump training courses must recognise the "Experienced Workers Route", that is, those with no formal training. Even those with an

NVQ Level 2 in Plumbing, the minimum formal industry qualification, will have no educational experience of design criteria such as heat loss calculations since they are not in the curriculum. Heat pump technology highlights the minimal approach to vocational education and training (VET) in the UK in comparison to the Continental emphasis on a more rounded educational demand for construction occupations that is apparent in the training manual for the EU Heat Pump Certificate, the EUCERT HP.

The thesis proposes that legislating technological change to impact on carbon dioxide emissions reduction needs to consider the interplay between legislation, technology and vocational education and training. It is thus an exploratory journey that combines a multidisciplinary approach to understanding the introduction of a new appliance into a market where existing technology has developed in conjunction with the VET qualities of the installer. The gas boiler, it could be argued, has evolved in response to the limitations of installer VET, its automation almost guaranteeing an acceptable minimum efficiency irrespective of system design. The thesis provides adequate evidence that, whilst heat pumps are sensitive in ways that fossil fuel appliances are not, heat pumps can undoubtedly work well and provide significant CO_2 savings when designed appropriately. The thesis suggests that a pre-requisite for this to occur is recognition of the designer role, both in its formal knowledge of engineering and its iterative decision-making faculties. The displacement of existing custom and practice requires a multi-faceted analysis of a range of technology-specific dependencies that include the implications of such policies on the central heating market, the manufacturers, the designers and installers.

Finally, Chapter 8 attempts to draw together the main findings of the thesis and explore their applicability to all low energy systems. The chapter outlines both these findings and the author's contribution to knowledge.

The issues raised in this introduction indicate that the analysis of the impact of any 'new' technology entering an established market requires an interdisciplinary approach that brings together a quantitative fact-based approach with the more nuanced advantages of qualitative exploration. The core objective of the RES and the EPBD Directives is the targeted reduction of CO_2 emissions through the support for new technologies including heat pumps where, for the residential market, this equates to the heat requirements for space heating and domestic hot water. Meeting these targets requires optimised heat pump output, thus we must consider the quality of design and installation since the premise of a low carbon future depends on the translation of a technical heating load assessment into a design specification and its correct realisation. Such a process must therefore include the vocational education and training experience of contractor and supply chain, a clear understanding of manufacturers' technical data and the ability to interpret, interpolate and iterate in the search for an optimal solution.

Chapter 1 Research context: Residential central heating - changing the status quo

Gas central heating

Some 89% of households in England (English Housing Survey Annex 3, 2010) have central heating with over 90% using natural gas (BRE, 2005). A similar density of central heating is found in Wales, 93% as of 2004 (Statistics for Wales 2004) and in Scotland, where the 'Poverty' website claims: "Fewer households now lack central heating in Scotland than Great Britain as a whole." The map, Figure 1-1, shows the proportion of properties without central heating and clearly shows that the lowest levels of central heating occur in rural areas, most likely off the national gas grid.

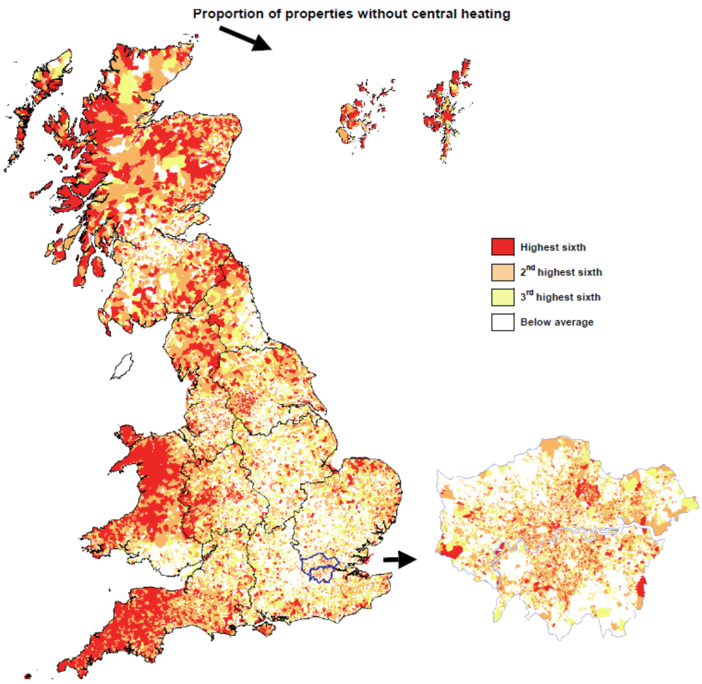
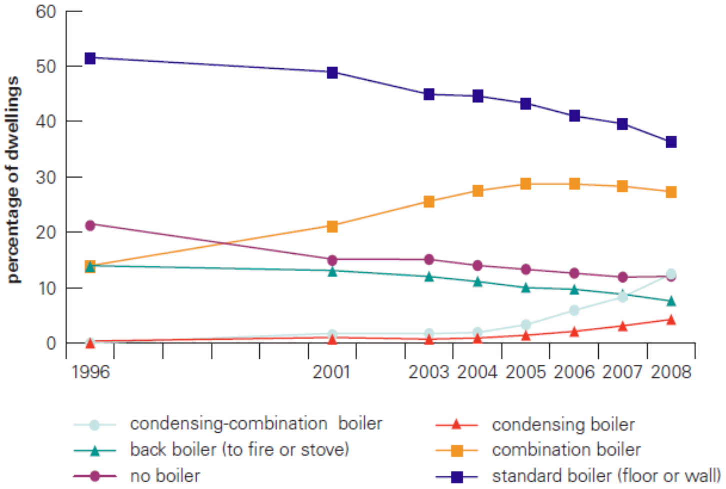


Figure 1—1 Proportion of households without central heating (the Poverty Site)

Condensing boilers are the most efficient boiler type and have been mandatory since 2005 for gas and 2007 for oil (ODPM, 2005). As of 2008, some 16-17% of all household boilers were condensing (English Housing survey Annex 6, 2010), whilst the condensing boiler market share reached 98% in June 2008, some 84% in SEDBUK Band A (Market update 2008). Since the early

1980s, installation of standard and back-boilers has been decreasing with the growing popularity of the combination boiler.



Base: all dwellings
Notes: underpinning data is presented in Annex Table 5
Source: English House Condition Survey 1996-2007, English Housing Survey 2008, dwelling sub-sample

Figure 1—2 Growth of condensing boiler installations (English Housing Survey, 2008)

The English Housing Survey, Headline Report 2008-09 (CLG 2010) presents the current situation for domestic heat sources, Figure 1-2, where only condensing models show growth as from around 2005, as would be expected in the light of the 2005 Amendments to Approved Document L (2002). New boilers must be selected based on SEDBUK efficiency bands A or B. SEDBUK (the Seasonal Efficiency of Domestic Boilers UK database) describes itself thus:

> "SEDBUK was developed under the Government's Energy Efficiency Best Practice Programme with the co-operation of boiler manufacturers, and provides a basis for fair comparison of the energy performance of different boilers. SEDBUK is the average annual efficiency achieved in typical domestic conditions, making reasonable assumptions about pattern of usage, climate, control, and other influences. It is calculated from the results of standard laboratory tests together with other important factors such as boiler type, ignition arrangement, internal store size, fuel used, and knowledge of the UK climate and typical domestic usage patterns. For estimating annual fuel costs SEDBUK is a better guide than laboratory test results alone. It can be applied to most gas and oil domestic boilers for which data is available from tests conducted to the relevant European standards. The SEDBUK method is used in SAP." (SEDBUK website)

SEDBUK is the UK response to the EU Council Directive 92/42/EEC of May 1992, on: "efficiency requirements for new hot-water boilers fired with liquid or gaseous fuels" which set out to promote energy efficiency and an internal EU market with various amendments and corrections made (European Commission 1992). The directive established a baseline for space heating boiler testing which provides a net calorific seasonal efficiency based on the average efficiency at 100% full and 30% part load. For SEDBUK purposes, net efficiency is converted to gross with separate conversion factors for condensing and non-condensing boilers (note the assumption of condensation occurring) and is further modified by particular boiler features such as regular or combi, on/off or modulating and permanent pilot or electronic ignition. Within the UK, The Government's Standard Assessment Procedure for Energy Rating of Dwellings, SAP2005 and SAP2009, (DECC 2011) provides the equations used to prepare the SEDBUK efficiency evaluation.

Originally, a SEDBUK rating did not apply to the production of domestic hot water, not even for combination boilers, and in recognition of this SAP2009 introduced winter and summer efficiencies to published SEDBUK values. The adjustments ($\Delta\eta_{winter}$ and $\Delta\eta_{Summer}$) only really impact on summer operation with a (-9.7)% reduction for regular boilers and (-9.2)% for combis.

In 2002 the EU responded to the need for an independent assessment of domestic hot water production through the promotion of measurement standards for hot water heaters, available through IEA Annex 28 (European Commission 2002). This has resulted in the SEDBUK database also providing "comparative hot water efficiency" ratings for boilers based on EN 13203: 2006, "Gas-fired domestic appliances producing hot water — Appliances not exceeding 70 kW heat input and 300 litres water storage capacity: Part 2 assessment of energy consumption," using the CEN Mandate 324 (EC, 2002) model of hot water draw-off described as "tapping pattern 2" [tapping cycle number 2] over 24 hours (Gastec personal correspondence). For high domestic hot water users, the lower comparative hot water efficiency measure is suggested.

These hot water efficiencies are considerably lower than seasonal space heating efficiencies, but for UK average central heating systems, annual domestic hot water represents perhaps only one third of the space heating energy (Highgate Society, Action Energy). The domestic hot water load is often of little significance in hard to heat, low SAP efficiency-band dwellings as evidenced by solar thermal installations with their minor impact on SAP rating and long payback periods. Conversely, Clarke (Clarke & Grant 2010) states that:

> "the total hot water demand [hot water and losses] of a [Passivhaus] hot water heating system is usually at least 30 kWh/m^2.a, double the Passivhaus heating demand of 15 kWh/m2.a."

Domestic hot water efficiency is critical for super-insulated buildings but of rather less importance for most poorly insulated properties."

SEDBUK Band A runs from 89% upwards (no boilers exceed 93%). Three significant UK field trials have tested condensing boilers, the Carbon Trust's 'Micro-CHP Accelerator' (Carbon Trust 2007), the Energy Savings Trust's (EST) 'In-situ monitoring of efficiencies of condensing boilers and use of secondary heating' (Orr, et al, 2009) and the EST's 'In-situ monitoring of efficiencies of condensing boilers –TPI control project extension' [*Time-Proportional Integral (TPI) Controls*] (Kershaw, et al, 2010). All three trials provide field mean efficiencies of about 85%. The Micro-CHP report comments:

> "These findings suggest that current installations of boilers in homes in the UK may frequently only achieve performance at a level around 4-5% below their SEDBUK declared efficiencies."

The EST 2009 executive summary comments:

> "The mean efficiency of the trial set of regular boilers [non combi] was 85.3% with a standard deviation of 2.5%. This is significantly less than that suggested by the mean SEDBUK seasonal efficiency of 90.4% (standard deviation 1.1%). Trial efficiencies can be directly compared to SEDBUK efficiencies, as data used to calculate efficiency was recorded at the boiler. To compare overall performance to combination boilers, performance of regular boilers should take into account losses from primary pipework and hot water cylinders. Tank and primary pipework losses were estimated from SAP at 900 kWh per year. Such a loss would reduce the effective overall efficiency of the boiler by about 5% (based on an annual heat output of 15,000 kWh). Thus a more valid mean regular boiler annual effective efficiency may be 80.3%" (Orr, et al, 2009, p*v*).

The 900 kWh/yr losses are based on averaging the losses for insulated primaries and cylinder (1200 kWh/yr) with older uninsulated primaries and poorly insulated cylinder (2500 kWh/yr) giving a mean of 1800 kWh/yr. It is then assumed that half of this is useful losses, in that it provides heat in winter, giving the 900 kWh/yr adopted by SAP. The report continues:

> "The simple "efficiency" of the boilers is calculated from heat out divided by the energy of the gas used (based on the higher [gross] calorific value) as a percentage. The simple efficiency (also referred to as the 'heat efficiency') is defined as:

$$\boldsymbol{Heat\ Efficiency}(\%) = \frac{(\boldsymbol{space+hot\ water})[\boldsymbol{kWh}]}{\boldsymbol{Gas\ burnt}[\boldsymbol{kWh}]} \times \mathbf{100} \qquad \text{Equation 1-1}$$

The SEDBUK database uses this definition of efficiency and where comparison is made between trial efficiency and SEDBUK values, the simple efficiency is used" (Orr, et al, 2009, p22).

It is clear from the Micro-CHP and both of the EST trials that this efficiency rating does not include electrical loads associated with controls, electronics, purging, fans, etc, often referred to as parasitic losses. The range of this electrical load varies with boiler make:

> "80% of boilers recorded annual electrical consumption greater than the SAP assumption of 175 kWh, ranging from around 100 kWh/year to over 750 kWh/year. There is a wide variation in boiler electrical consumption between installations supplying similar amounts of heat. Detailed analysis of electrical and gas consumption of boilers indicated that a key factor in electrical consumption is the pump operating hours/month which is, in turn, dependent upon the setting of the room thermostat, TRVs and other controls." (Orr, et al, 2009, p84).

Importantly, no values are given for efficiency based on kWh of heat out over kWh of gas and electricity in:

$$\boldsymbol{\eta_{th}} = \frac{\boldsymbol{Heat_{out}}}{\boldsymbol{Gas_{in}+Elec_{in}}}$$ Equation 1-2

In recognition of the need to refine efficiency for CHP units, the authors of the Micro-CHP Accelerator report introduce the concept of Carbon Benefit Ratio (CBR) to measure the carbon dioxide impact of CHP but also apply it to condensing boilers, as do the EST condensing boiler trials. The EST Report (Orr, et al, 2009) comments that CBR can also be use to compare performance of micro-CHP and boilers with heat pumps. CBR is defined as:

$$\boldsymbol{CBR}(\%) = \frac{\left(\boldsymbol{useful\ heat\ delivered}\ [\boldsymbol{kWh}]\times\boldsymbol{CEFgas}\left[\frac{\boldsymbol{kgCO2}}{\boldsymbol{kWh}}\right]\right)}{\left(\boldsymbol{gas\ burn}\ [\boldsymbol{kWh}]\times\boldsymbol{CEFgas}\left[\frac{\boldsymbol{kgCO2}}{\boldsymbol{kWh}}\right]\right)+\left(\boldsymbol{electricity\ used}\ [\boldsymbol{kWh}]\times\boldsymbol{CEFelectricity}\left[\frac{\boldsymbol{kgCO2}}{\boldsymbol{kWh}}\right]\right)}$$

Equation 1-3

where CEF represents the carbon emissions factor for the fuel and electricity.

Electricity used by the pump, fan and controls, appears in various proportions either as useful space heat or useful DHW. Not all of this will be accounted for by heat meters in the primary circuit. Directly measured casing losses would capture most of the stray heat from electricity in the case of system boilers (those that contain all electrical components for operation), but not where the circulation pump is somewhere else. For regular condensing boilers (non-combi), mean annual electrical loads appear to be around 200 kWh (Orr, et al, 2009, p43), although the TPI trial states:

> "[At] very low heat loads the start up losses have a more dominant influence on the overall system efficiency and therefore result in a disproportionately poor CBR%" (Kershaw, et al, 2010, p24). Unfortunately, there does not appear to be widely available data on CBR for the range of heating options and thus its use is limited as a practical measure of system efficiency.

Since the gas condensing boiler is the *de facto* prime choice for domestic central heating, the EST condensing boiler field trials provide a current benchmark for alternative central heating heat sources where mean thermal efficiency for boiler and cylinder can be taken as approximately 85% with a reduction to 80% where all electrical loads are considered – including, importantly, the system pump. Since all wet central heating systems have a pump, one could argue that whilst the boiler control electronics should legitimately be included in boiler efficiency calculations, circulation pump loads are dependent on pump mass flow rate and index circuit resistance and are thus variable for the same boiler. We need to establish system 'boundaries' for efficiency analysis if we wish to compare the efficiency of one source to another.

Defra provide carbon dioxide conversion factors for all fuels (DEFRA 2011). Emissions may be classified as total greenhouse gas emissions associated with supply and consumption, known as "all scopes". Published values for natural gas and grid supplied electricity are:

> Natural Gas (gross calorific value): 0.20155 kgCO_2/kWh total CO_2 equivalent
> Electricity "CONSUMED": 0.59368 kgCO_2/kWh total CO_2 equivalent

A single definitive allowance for boiler electricity is fraught with ambiguity since the mean annual load of 200 kWh quoted from the EST Report (Orr, et al, 2009) includes the circulation pump yet all wet heating systems, including those with heat pumps, have such a pump. The electrical load from the appliance fan, valves and electronics is not provided. This fraction of the total electrical load could also be similar to that of a heat pump since both have internal controls and valves. Thus, for simplicity, a condensing boiler operating at between 85% or 80% efficiencies, the CO_2 equivalence lies between (0.202/0.85) to (0.202/0.80) giving 0.237 to 0.252 kgCO_2/kWh. Where a dwelling is on the natural gas grid, any alternative heating appliance must minimally sit in this emissions band in order to not increase CO_2 emissions. Heat pumps offer an alternative to the gas condensing boiler but since they are driven by electricity, their minimum seasonal efficiency at today's electricity grid fuel mix needs to be between (0.594/0.252) and (0.594/0.237), approximately 236% to 250% or 2.36 to 2.50 just to match the emissions from gas at 80% and 85% respectively; there are, at these efficiencies, currently no carbon savings.

Carbon dioxide emissions reduction

In the UK, CO_2 emissions from dwellings amount to some 27% of the national total with approximately three quarters of emissions from space heating and hot water, Figure 1-3; a similar situation occurs in many of the EU member states.

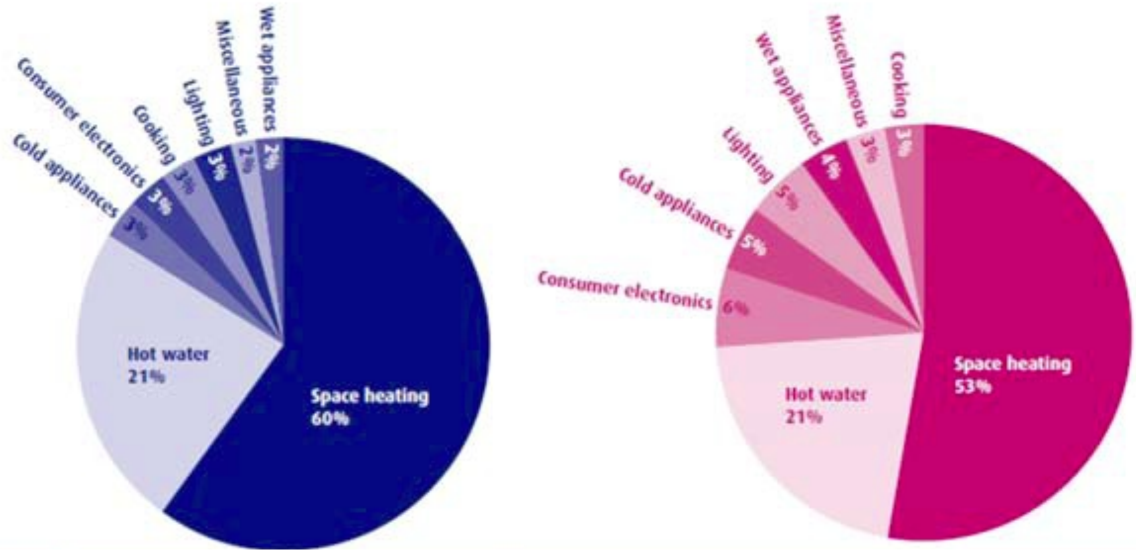


Figure 1—3 Energy and CO2 emissions from dwellings (UK DTI Energy consumption tables 2004)

The IPCC identify the buildings sector as offering the largest low cost potential CO_2 savings in all world regions by 2030 (IPCC Fourth Assessment Report), with Chapter 6, "Residential and commercial buildings" (Levine, et al, 2007**)** outlining specific reduction measures including the use of heat pumps (see for example pages 394, 397 and 402). Whilst there has been some criticism of the report methodology, see for example Lowe (2007), both EU and member state policy has been developed reflecting the IPCC viewpoint.

According to the European Commission Climate Action website:

> "In March 2007 the EU's leaders endorsed an integrated approach to climate and energy policy that aims to combat climate change and increase the EU's energy security while strengthening its competitiveness. They committed Europe to transforming itself into a highly energy-efficient, low carbon economy. To kick-start this process, the EU Heads of State and Government set a series of demanding climate and energy targets to be met by 2020, known as the "20-20-20" targets. These are:

- A reduction in EU greenhouse gas emissions of at least 20% below 1990 levels
- 20% of EU energy consumption to come from renewable resources
- A 20% reduction in primary energy use compared with projected levels, to be achieved by improving energy efficiency.

The EU leaders also offered to increase the EU's emissions reduction to 30%, on condition that other major emitting countries in the developed and developing worlds commit to do their fair share under a global climate agreement."

On the 26 of May 2010, the European Commission published a communication: "Analysis of options to move beyond 20% greenhouse gas emission reductions and assessing the risk of carbon leakage," which revisits the implications of 20% and 30% target ambitions (European Commission, 2010).

As a member state, the UK Coalition government has responded by publishing a low carbon transition plan, "The Carbon Plan" (DECC 2011) along with a programme which outlines pathways to an 80% reduction in greenhouse gas emissions by 2050: "Planning our electric future: a white paper for secure, affordable and low carbon electricity" (DECC 2011). With UK current emissions for electricity generation around 500 grams CO_2/kWh, the Coalition government proposes a 2030 target of 50 grams CO_2/kWh to be partly achieved by applying feed-in tariff policies, see Figure 1-4.

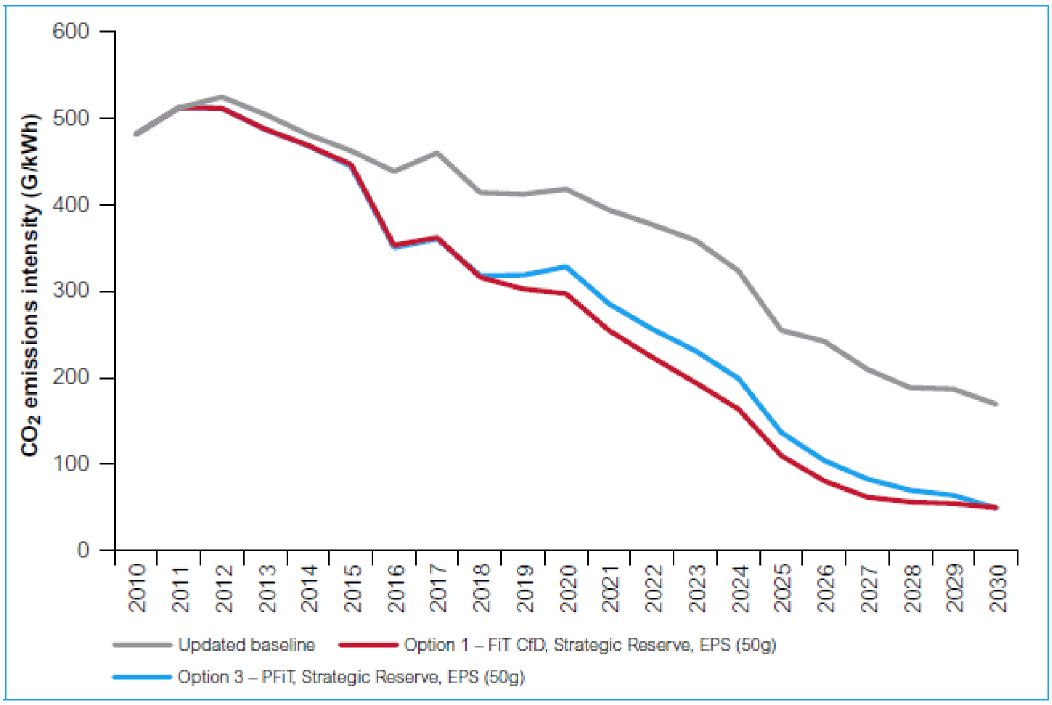


Figure 1—4 UK projected CO_2 from electricity generation up to 2030 (DECC 2011, p114)

The role of heat pumps

At the European Union level, the European Commission passed Directive 2009/28/EC of the European Parliament and Council of 23 April 2009 on the "Promotion of the use of energy from renewable sources", the RES Directive (European Commission 2009). The Directive calls for, *inter alia*, "a 20% target for the overall share of energy from renewable sources". The RES specifically identifies heat pumps in its Annex VII.

Within the UK, both DECC and the Committee on Climate Change (CCC, 2010) foresee a key role for heat pumps in enabling the UK to fulfil its climate and energy goals. Among its many references to heat pumps the CCC report states:

> "Key technologies which should be demonstrated now for deployment in the 2020s include CCS [carbon capture and storage] in power generation and industry, electric cars and vans, and electric heat pumps," p13.
>
> "The key option for supply-side decarbonisation in this scenario is heat pumps," p24.
>
> "Buildings: Direct emissions from heat in buildings are reduced significantly by 2030, as a result of major improvements in energy efficiency and roll-out of low-carbon heat, especially heat pumps," p29.

The UK Government's more recent document, "The Future of Heating: A strategic framework for low carbon heat in the UK" (DECC 2012), similarly enthuses over heat pumps:

> "As the electricity system decarbonises, technologies such as heat pumps and even electric resistive heating in buildings will be an increasingly effective way to decarbonise heat supply," p18.
>
> "Many new homes are now fitted with a heat pump, *able to operate three to four times more efficiently than a gas boiler*, and businesses are increasingly using heat pumps as a convenient way to both heat and cool their buildings," p39 (author's italics).
>
> "[H]eating technologies that use low carbon electricity hold particular promise, especially as electricity is universally available and technologies here are relatively established. In addition, the high efficiencies of heat pumps, combined with improved building and storage technologies, could counteract the relatively high costs of electricity, making electrical heating an affordable option, particularly if the

manufacturing and installation costs of heat pumps come down as volumes increase," p44.

"In suburban and rural areas, in particular, low carbon heating technologies at the level of individual buildings will be necessary. Here, heat pumps are expected to provide substantial quantities of heat where heat networks … are not technically or economically viable," p56.

"The Government's vision is of buildings benefitting from a combination of renewable heat in individual buildings, particularly heat pumps," p93.

Heat Pumps and Renewable Heat Incentives

Preparatory work on implementing a renewable heat incentive was developed under the previous Labour administration with DECC publishing "The UK Supply Curve for Renewable Heat. Study for the Department of Energy and Climate Change." (DECC 2009). The report proposed the market development of both air and ground source heat pumps. The UK Coalition government, which came into power in 2010, has broadly followed the Supply Curve advice on renewable heat and, complimenting its approach of applying feed-in tariffs, has responded with the "Renewable Heat Initiative"(RHI) (DECC 2011):

> "We aim to launch the Renewable Heat Premium Payments in July 2011 and will announce further details in May 2011. A second phase of RHI support that will include long-term tariff support for the domestic sector will then be introduced in 2012 to coincide with the introduction of the Green Deal for Homes."

The government proposed to initially support only ground source heat pumps:

> "Air source heat pumps will not be supported from the outset because more work is needed to better understand the costs associated with the technology and, for air to air heat pumps, work is ongoing to develop a robust methodology for measuring heat delivered in the form of hot air. Subject to successful conclusion of this work and other factors (such as the role of cooling as opposed to heating in such systems) we intend to extend eligibility to this technology from 2012."

The RHI document states that:

> "The EU standard, given in *Annex* VII of the RED [Renewable Energy Directive] is based on the total useable heat delivered, the average seasonal performance factor and the efficiency of electrical generation. The Commission has committed to providing

guidance on how these factors should be measured and we may review our approach once the Commission issues this guidance. To avoid introducing a potentially complex system in advance of the Commission's guidelines, rather than referring to usable heat or seasonal performance, the RHI will require a COP of 2.9. Applicants will be required to demonstrate, to Ofgem's satisfaction, that the heat pump meets a COP of at least 2.9; this will usually be part of the equipment documentation supplied by the manufacturer," p36.

This is an important clause, referring to footnotes which state:

"All heat pumps have a "coefficient of performance" (COP), defined as the ratio of the amount of heat output per unit of energy input. The MCS [Microgeneration Certification Scheme] requires COP greater than 2.9. The method for measuring COP is set out within the MCS standards. Any MCS equivalent products will also be required to achieve a COP of 2.9 or above."

The MCS Product Certification Scheme for heat pumps, MCS 007 (MCS, 2009, pp7-8), provides the following guidelines:

"For compliance with this scheme, heat pumps must be optimized for heating and must achieve the following minimum COP when tested in accordance with EN 14511-3-2004 at EN 14511-2-2004 rating conditions."

The relevant heat pump types and testing conditions are shown in Table 1-1. It is clear that a ground source heat pump COP of 2.9 is below that required in the current documentation.

Heat Pump type	Min COP	EN 14511-2 Rating Conditions	
Ground/Water	3.5	Table 7	Standard Rating conditions – Brine (for floor heating or similar application)
Water/Water	3.8	Table 7	Standard Rating conditions – Water (for floor heating or similar application)
Air/Water	3.2	Table 9	Standard Rating conditions – Outdoor air (for floor heating or similar application)

Table 1—1 Based on MCS Test requirements for COP (MCS, 2009)

As will become apparent in Chapter 3, the efficiency of a heat pump is maximised at minimum source/sink temperature difference. EN 14511-2-2004 tables 7 and 9 show minimum COP based "standard rating conditions" to refer to underfloor heating, Table 1-2, rather than the more typical high temperature radiator systems common to the UK.

		Outdoor heat exchanger		Indoor heat exchanger	
EN 14511 Test	Type	Inlet Temperature °C	Inlet Temperature °C	Inlet Temperature °C	Outlet Temperature °C
Standard rating conditions	Ground/Water	0	(-3)	30	35
Standard rating conditions	Water/Water	10	7	30	35
		Inlet dry bulb Temperature °C	Inlet wet bulb Temperature °C	Inlet Temperature °C	Outlet Temperature °C
Standard rating conditions	Air/Water	7	6	30	35

Table 1—2 Heat pump 'standard rating conditions', based on EN 14511-2-2004

The Renewable Energy Directive specifically addresses minimum heat pump efficiency in Annex VII (European Commission 2009):

> "The amount of aerothermal, geothermal or hydrothermal energy captured by heat pumps to be considered energy from renewable sources for the purposes of this Directive, *ERES*, shall be calculated in accordance with the Equation 1-4:
>
> $$\boldsymbol{ERES} = \boldsymbol{Q_{usable}} \times (\mathbf{1} - \mathbf{1}/\boldsymbol{SPF}) \qquad \textbf{Equation 1-4}$$
>
> where:
>
> Q_{usable} = the estimated total usable heat delivered by heat pumps fulfilling the criteria referred to in Article 5(4), implemented as follows: "Only heat pumps for which $SPF > 1{,}15 * 1/\eta$ shall be taken into account,
>
> SPF = the estimated average seasonal performance factor for those heat pumps,
>
> η is the ratio between total gross production of electricity and the primary energy consumption for electricity production and shall be calculated as an EU average based on Eurostat data.
>
> By 1 January 2013, the Commission shall establish guidelines on how Member States are to estimate the values of Q_{usable} and SPF for the different heat pump technologies and applications, taking into consideration differences in climatic conditions, especially very cold climates."

This minimum seasonal efficiency at the relevant system boundary has been resolved by the European Commissioners' Decision of March 2013 (EC Decision, 2013) and depends on the climate zone for the installation. For the UK there had been no such decision at the time of the Energy Savings Trust Heat Pump trial design, pre-2009, or when MCS standards were being drafted and presented by BERR (BERR, 2008).

There remain a number of complex issues in Annex VII which will require resolution, not least of which is the gathering of statistical data on heat pump seasonal performance from each member state. The minimum value of SPF will decrease as renewable energy begins to impact on the grid generation efficiency (η or *eta*), see Table 1-3. In lieu of further guidance from the Commission on SPF, the UK government set the minimum ground source heat pump COP of 2.9, which matches the electricity ratio at 39.96 or 40%. Eurostat provide values for η (eta), the ratio of total gross production of electricity to the primary energy consumption for electricity production for the EU27, ranging from 40.49% in 1990 to 45.35% for 2011 (Eurostat, 2013). The impact of eta on minimum SPF is shown in Table 1-3.

Year	eta	1/eta	SPF min
1990	0.4049	2.47	2.84
1995	0.4129	2.42	2.79
2000	0.4277	2.34	2.69
2005	0.4399	2.27	2.61
2010	0.4571	2.19	2.52
2011	0.4535	2.21	2.54
Future?	0.5	2.00	2.3

Table 1—3 Eta values for SPF > 1.15 x 1/η (based on Eurostat values, 2013)

Renewable Heat Premium Payment scheme

Whilst the Renewable Heat Incentive is still to be launched (February, 2013), the Renewable Heat Premium Payment scheme (DECC RHPP, online) has been introduced where, rather than continuous payments for renewable heat, the installation is supported by a grant of £1,300 for air to water and £2,300 for ground-to-water heat pumps.

UK heat pump sales have been subject to continuous subsidy through various government grant mechanisms over the last decade or so including the Blue Skies grant, the Low Carbon Building Programme and currently the RHPP. Sales remain volatile as exemplified by the two peaks in registered installations in March 2012 and March 2013 coinciding with the announced end of phases 1 and 2 of the Renewable Heat Premium Payment scheme, Figure 1-5. The long-term impact of such subsidies on UK carbon dioxide emissions is hard to determine for, in comparison to boilers, heat pumps are currently of minor significance in the UK domestic heating market, their presence, it would appear, almost entirely dependent on Government subsidy.

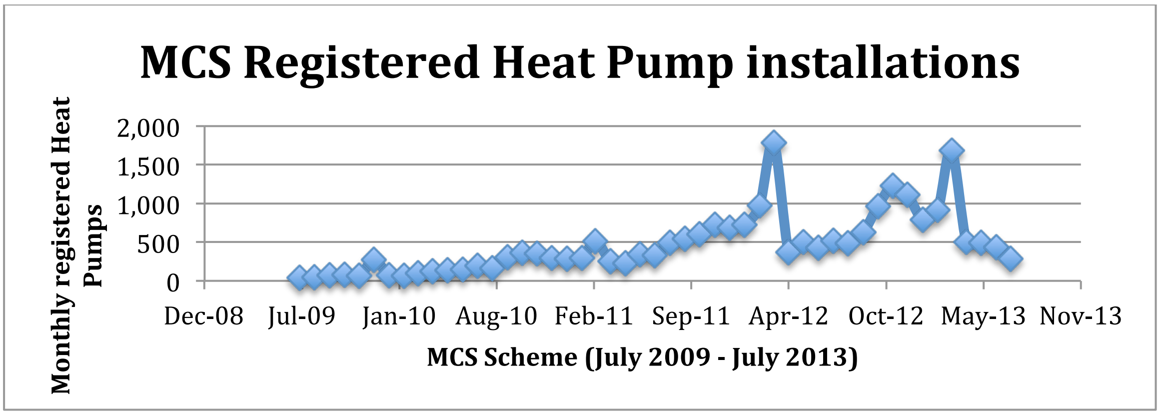


Figure 1—5 MCS registered monthly installations since July 2009 (based on MCS Installation Statistics)

The MCS scheme is limited to heat pumps up to 45 kW (MCS 2013), reflecting its primary scope to be dwellings. Boilers dominate the UK central heating market with annual reported sales to be some 1.59 million in 2008 (Market Update, 2008). In comparison, the European Heat Pump Association present the figure of 21,360 for UK heat pump sales in 2011 (Nowak 2012). Surprisingly, if Nowak's figures are correct, only 5,314 were registered with MCS (MCS installation statistics) making them eligible for grants since all RHPP installations must be registered with the Microgeneration scheme. The aim of the RHPP is to monitor up to 700 heat pump installations (Personal correspondence, Chris Wickins, Principal engineer, RHI team) to assess their seasonal performance factor, their contribution to carbon dioxide emissions reduction and evidence for setting the RHI tariff rates.

Defining Seasonal Performance Factor

The International Energy Agency's (IEA) Annex 28 programme was set up to ensure comparability between heat pumps and other forms of heating for compliance with the Energy Performance of Buildings Directive, 2003, the EPBD. The programme appears to be based on original work for the Swiss Federal Office of Energy by Afjei and Wemhöner: "Seasonal performance calculation for residential heat pumps with combined space heating and domestic hot water production" (Afjei, 2002). The results: "Standardised testing and seasonal performance calculation for multifunctional heat pump systems" (Wemhöner et al 2007 p6) provide two definitions for heat pump seasonal performance factor based on the position of the system boundary, SPF_{hp} and SPF_g:

> "The heat pump SPF (SPF_{hp}) corresponds to the system boundary used in European heat pump testing according to EN 14511 [the heat pump alone] and characterises the heat pump operation. The generator SPF (SPF_g) is the ratio of the produced energy of all generators (in this case heat pump and electrical back-up) to the respective electrical

energy input *and is well suited for the comparison to other heat generators like boilers"* [author's italics].

Annex 28 provides a theoretical procedure to identify SPF using weather data, based on the "bin method". Weather bands are centred on EN 14511 COP test results with the SPF calculated as a weighted mean. Wemhoner and Afjai have been perhaps the leading researchers for heat pump seasonal efficiency, whose work appears to have set the foundations for SPF calculations for air, ground, exhaust and compact service units (a heat pump driven MVHR and DHW unit popularised by the *Passivhaus Institute*). Key to their work has been the establishment of appropriate system boundaries that provide transparent comparisons between system typologies and different heat sources. Their 'Final Report' for the IEA HPP Annex 28 "Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating. Part 1: Proposals for calculation method and test procedure" (Wemhoner & Afjai 2006), will be considered in greater detail in Chapter 4.

SPF boundary definitions have been developed by the S P Technical Research Institute, Sweden, to produce: "Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning" (Nordman, et al, 2010), where four levels of SPF are identified, Figure 1-6.

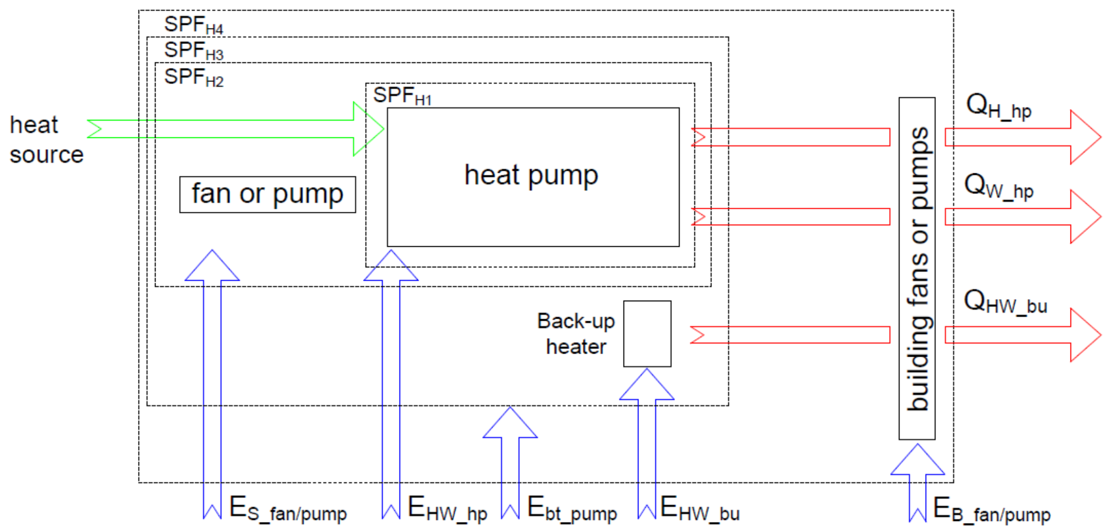


Figure 1—6 SPF system boundaries (Nordman, et al, 2010)

The Renewable Energy Sources Directive, 2009, did not define SPF nor did it reference an SPF calculation to be used, however, in pursuit of the relevant boundary, the Commission has undertaken heat pump trials through the 'Intelligent Energy Europe' research programme:

"SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector" (SEPEMO). SEPEMO has set the followings aims:

- Guideline for measuring seasonal performance of heat pump systems in the most common configurations which are used in Europe including the documentation of the necessary sensors and a definition of all storage parameters. Additional parameters to clarify the boundary conditions under which the system is running.
- A defined methodology for calculation of the seasonal performance factor (SPF) and a definition of which devices of the system have to be included in this calculation.
- Evaluation method for comparison of the heat pump system with conventional heating systems.
- Method for benchmarking the seasonal performance under consideration of the boundary conditions under which the system is running.

SEPEMO, lead by Nordman, have adopted the boundaries shown in Figure 1-6. SEPEMO monitoring guidance demands both SPF_{H2} and SPF_{H3} measurements and the online trial database (http://www.sepemo.eu/field-test-sites/) provides results in SPF_{H3}, that is, inputs from source fan/pump, compressor and any backup.

As we have seen, for the purposes of the RES 2009, the issue of the measurement boundary was resolved in the EC Decision of March 2013 with the relevant boundary set as SPF_{H2} or, where monitoring is not available, results from "Testing and rating at part load conditions and calculation of seasonal performance", known as "SCOPnet" (BS EN 14825:2012). The EC Decision provides air-to-water and ground-to-water SPF_{H2} values based on three climate zones, cold, average and warm. All zones require a minimum SPF_{H2} of 3.5 for ground source whereas air source units have SPF_{H2} values of 2.5, 2.6 and 2.7 respectively. The EC Decision splits the UK into two climate zones, the actual renewable heat value for air source will depend on geographical location. DECC, in their latest review of the EST heat pump field trials (Dunbabbin, et al, 2013), consider that all heat pumps operating an SPF_{H2} of greater than 2.5 provide renewable heat, the value identified earlier in the chapter from CO_2 emissions and based on equivalence with condensing gas boilers at 85% efficiency. This SPF_{H2} definition of renewable heat obscures the role of any backup such as electrical resistance heating, which lies within the SPF_{H3} boundary.

It is apparent from this review of the relevant literature that there was and still is some confusion over the minimum performance to justify heat pumps as a lower carbon alternative to gas central heating.

EST Heat Pump Trials, 2009-2010

It was within this environment of EU Directives, tentative responses from member states and on-going European heat pump trials that the Energy Savings Trust began its UK heat pump field trials in 2009. The results from the first year of the trials, “Getting Warmer, a field trial of Heat Pumps” (EST 2010), were generally poorly received by the national and trade media whilst the EST and manufacturers tended to blame the installers for poor heat pump selection (matching pump to system) and, in general, a low standard of installation. The report could be considered as an optimistic take on the trial results which show low mean values of COP and entire system efficiency:

> “The performance values we monitored in the sample heat pumps varied widely; the best performing systems show that well-designed and installed heat pumps can operate well in the UK.” (EST 2010 p6)

The report defines coefficient of performance and system efficiency as follows:

> “Coefficient of performance (COP): The amount of heat the heat pump produces compared to the total amount of electricity needed to run it. The higher the COP, the less electrical energy is required to deliver a given amount of heat: a high COP shows good performance, and a low COP shows poor performance.
> System efficiency: The amount of heat the heat pump produces compared to the amount of electricity needed to run the entire heating system (including domestic hot water; supplementary heating; and pumps). This report’s conclusions and recommendations are based on the measured system efficiency.” (EST 2010 p7)

The report does not reference “system efficiency” against any particular measurement methodology although it does state that the results were:

> “peer reviewed by leading EU heat pump experts, including the SP Technical Research Institute of Sweden, Planair (Switzerland) and Germany’s Fraunhofer Institute, as well as UK stakeholders including the Energy Technologies Institute (ETI).” (EST 2010 p5)

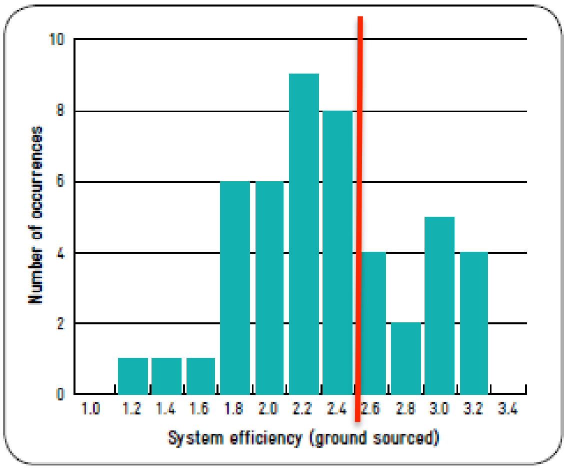
SP Technical are the principal investigator for the SEPEMO trials, Planair are a lead investigator for IEA heat pump Annex 37 “Demonstration of field measurements of heat pump systems in buildings - Good examples with modern technology”, Fraunhofer have completed their own German field trials and also contribute to SEPEMO, and the ETI are currently trialling heat pumps in the UK. See Chapter 5 for further discussion of European field trials.

An overview of the EST field trial results is shown in Figure 1-7. The report gives no further details on COP results other than those shown in Figure 1-7.

	Heat pump COP		System efficiency	
	Air source	Ground source	Air source	Ground source
Range	1.2 - 3.3	1.3 - 3.6	1.2 - 3.2[1]	1.3 - 3.3

Figure 1—7 COP and System Efficiency (EST 2010 p16)

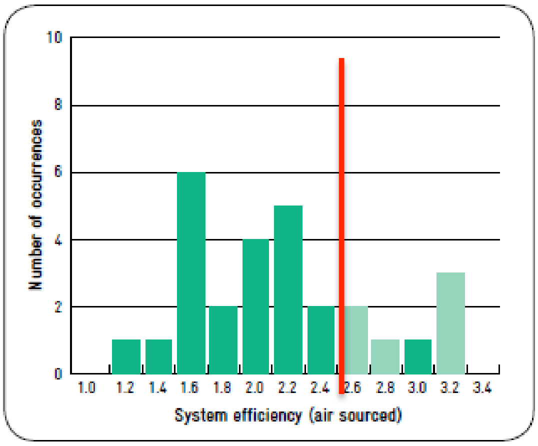
Histograms from the trial for system efficiency are shown in Figure 1-8 (ground source) and Figure 1-9 (air source).



Distribution of measured system efficiencies (GSHP)

Figure 1—8 Ground source heat pump "system efficiency" where 31% ≥ 2.5 (EST 2010 p15)

Performance figures for 47 ground source heat pumps are shown in Figure 1-8, out of which 15 (31%) have seasonal efficiencies greater than 2.5, or better than an 85% efficient gas boiler system. Conversely, nearly 70% produced more CO_2 than a condensing gas boiler.



Distribution of measured system efficiencies (ASHP)
Shaded bars denote estimated efficiencies

Figure 1—9. Air source heat pump "system efficiency" where 25% ≥ 2.5 (EST 2010 p15)

Of the 28 air source heat pumps, 6 out of the 7 top performing installations are estimated efficiencies, Figure 1-9. Even when these estimated efficiencies are included, only 25% produce less CO_2 than a gas condensing boiler.

Among the "Key Findings", the report states the following:

> "The major difference between the UK and European field trial findings is that the UK has particularly old and inefficient housing stock. The British climate also tends to be cold and damp, rather than very cold and dry like Scandinavia. But as well as these predictable factors, heat pump performance is affected by the existing heating systems in UK homes, the attitudes and behaviour of heat pump users, and the quality of installations. The heat pump market is more mature in the rest of Europe, and installers have more experience." (EST 2010 p15)

Where the national media picked up on the trials, they took an altogether more pessimistic view, typical of which is the Guardian (Vaughan, 2010) "UK heat pumps fail as green devices, finds study", which states that:

> "The Trust blamed the use of multiple contractors for fitting systems instead of a single contractor as used in Europe, wrongly sized systems, complicated controls and a lack of education for householders using them."

The Daily Telegraph (Morgan, 2010) in a critique of the RHI:

> "The Daily Telegraph understands that new unpublished trials conducted by a government-funded body show that the heat pumps are only "variably" effective at heating homes…. A spokesman for the EST declined to comment on the results, but said: 'Our responsibility to the public and the industry means we need to be 100% sure that

the data have been fully understood and cross-referenced before this can be publicly released. We intend to make the results of the first phase of work public before autumn. Out of 83 sites monitored across the UK during the trial, results indicated wide-ranging performance. We are aiming to secure funding for a second phase of the trial, so we can work out what is causing this variation, focusing on exactly what determines a high-performing heat pump retrofit installation, and ensure this becomes standard practice'."

The Delta Whitepaper, published through the SEPEMO website, "Heat Pumps in the UK: How Hot Can They Get?" (Delta Energy and Environment 2011) is, as would be expected, far more circumspect in its response to the EST trials. The Delta report compares UK field trial results with trials for both air and ground source heat pumps monitored in Germany and Switzerland and notes:

"That due to *differences in methodology between trials, the results are not completely comparable* [author's italics] – due to the wider system boundary used in the EST trial, the UK results are likely to be lower (possibly by a SPF of around 0.1) than the other trials.

Other important differences to note, which may contribute to lower SPFs in the UK, are:

1. The German and Swiss heating systems are typically of higher quality than those in the UK (in terms of the quality of components and control system).

2. UK and German installations were providing a higher proportion of DHW than in Switzerland.

3. UK buildings were (broadly) of lower quality in terms of insulation / rate of heat loss.

These issues may have reduced achievable SPF in the UK by a few percentage points, but these factors alone are not sufficient to explain the UK trial's poorer results."

With regard to point 3:

It is not obvious which way low levels of insulation would push the results. A high heat loss building could be re-modelled as a larger building with low heat loss in which case the heat loss parameter (W/m^2K) is of secondary importance to the heat pump performance. Given high performance space heating systems, a higher ratio of space to domestic hot water heat would increase combined SPF (space + water), by lowering the output temperature required of the heat pump. So poor envelope performance could be good for SPF. However, in the case of poorly performing space heating systems, poor envelope performance would be bad for SPF.

The Delta report concludes:

> "In Delta's view the gaps between UK and Swiss & German heat pump performance can be closed if there is a concerted effort – led by the heat pump industry - focusing on setting guidelines/standards for training and skills, and putting in place the framework to build skills of installers. A number of UK heat pump players are already making such efforts."

European trials will be further considered in Chapter 5 along with the "differences in methodology" and a detailed analysis of the EST trials in Chapter 6.

Delta's optimistic view on "closing the performance gap" is shared by the UK Committee on Climate Change. The Fourth Carbon Budget - reducing emissions through the 2020s (CCC 2010) makes specific reference to the EST heat pump trials:

> "For this analysis, it has been assumed that COPs start from current levels of 2.0 to 2.5. They are projected to increase towards an eventual plateau in the 2020s, with space heating COPs in the range 3.5 - 5.5 (up to 4.5 in residential applications and 5.5 in non-residential).
>
> The Energy Saving Trust (EST) recently published the results of the first large scale trial of heat pumps at 83 sites in the UK. A key finding was that heat pump performance can vary considerably between installations, and is particularly sensitive to installation and commissioning practices and customer behaviour.
>
> In the trials, GSHPs had a mid range of around 2.3-2.5, with the highest values above 3.0. The mid range of COPs for ASHPs was around 2.2, with the highest values over 3.
>
> The results of the EST field trial have important implications for the roll out of heat pumps in the UK:
>
> - In general, well installed and operated heat pumps are a suitable technology for reducing emissions in the UK.
> - Given the sensitivity of performance to design and commissioning, there is a requirement for improved training for installers.
> - Many customers expressed difficulty understanding the instructions, and this underlines the importance of improved information provision and technical support." (CCC 2010 p207)

Achieving a "COP" of up to 4.5 in residential property would require about a doubling of the field trial mean for ground source heat pumps.

The EST UK heat pump trials sit in an historical continuum of field trial research where a tried and tested methodology is applied to a new sphere of technology in order to assess its "real world", operational efficiency. Such a programme fits into a 'technology readiness level' (TRL) methodology, developed originally by NASA as a nine point check-list for technology innovation from theory to full application and since applied across a range of scientific, governmental and manufacturing processes. The TRL concept is a tool in 'technology readiness assessment' (TRA), described by the US Department for Homeland Security (Department of Homeland Security Science and Technology Directorate, 2009) as: "A systematic, metrics-based process and resulting report that assesses the maturity of technologies." The Homeland Security method comprises three assessments: the technology readiness level, the manufacturing readiness level and the programmatic readiness level, although the three categories are in practice not discrete but interwoven with feedback loops.

The TRL process continues to evolve:

> "There has also been a proliferation of TRL offshoots, including "Design Readiness Levels"; "Material Readiness Levels"; "Manufacturing Readiness Levels"; "Integration Readiness Levels"; "Innovation Readiness Levels"; "Capability Readiness Levels"; *ad infinitum*– a process that can be continued to a *reductio ad absurdum*! That being said, all of these offshoots reflect recognition that we are not doing well in the process of developing and infusing technology, and that there are various aspects to the process that must be dealt with in a more propitious manner." (JB Consulting International).

JB Consulting provide extended definitions, "tailored by individual projects within the context of the basic TRL descriptions defined by NASA 1995" where, having moved beyond prototype testing at stage 7, stages 8 and 9 provide the following, Table 1-4:

Level	**NASA (2007)**	**DOD (2005)**	**NATO**	**UK MOD**
8	The final product in its final configuration is successfully demonstrated through test and analysis for its intended operational environment and platform (ground, airborne or space).	Technology has been proven to work in its final form and under expected conditions. In almost all cases, this TRL represents the end of true system development. Examples include developmental test and evaluation of the system in its intended weapon system to determine if it meets	Technology has been proven to work in its final form and under expected conditions. In almost all cases, this TRL represents the end of demonstration. Examples include test and evaluation of the system in its intended weapon system to determine if it meets design specifications, including those relating to supportability. Not R&T	Tailored by individual projects within the context of the basic TRL descriptions defined by NASA 1995

		design specifications.	funded although R&T experts may well be involved.	
9	The final product is successfully operated in an actual mission.	Actual application of the technology in its final form and under mission conditions, such as those encountered in operational test and evaluation. Examples include using the system under operational mission conditions.	Application of the technology in its final form and under mission conditions, such as those encountered in operational test and evaluation and reliability trials. Examples include using the final system under operational mission conditions.	Tailored by individual projects within the context of the basic TRL descriptions defined by NASA 1995

Table 1—4 Technology Readiness Levels (JB Consulting International)

Within the context of such an approach, the Europe-wide SEPEMO field trials are aimed at providing definitive operational data for Stage 8 assessment before the European Commissioners pronounced on minimum heat pump efficacy for reducing carbon dioxide emissions, to clear them for "mission readiness".

Chapter 2 Methodology

The issues raised in this introduction indicate that that analysis of the impact of any 'new' technology entering an established market could be supported by an interdisciplinary approach that brings together the quantitative, based on heat pump trial operation data, with the advantages of qualitative exploration.

Research Question

The key questions that arise from the review is:

How well do heat pump systems work in practice? To what extent are heat pump field trial results being communicated in a consistent fashion and what lessons can be learned from heat pump field trials?

Data uncertainty

The first three chapters provide market context, thermodynamic fundamentals and testing regimes (including a pilot field study) necessary for understanding and analysing field trial outputs. These outputs are based on trial reports and raw data files. Formal acceptance of published trial data should require some knowledge of its 'uncertainty', a combination of accuracy and precision. The European Commission's guidance on acceptance of data uncertainty (EC, 2012) provides the following description:

> "When determining the quality of measurements, international standards refer to the quantity of "uncertainty". This concept needs some explanation. There are different terms frequently used in a similar way as uncertainty. However, these are not synonyms, but have their own defined meaning:
>
> - Accuracy: This means closeness of agreement between a measured value and the true value of a quantity. If a measurement is accurate, the average of the measurement results is close to the "true" value (which may be e.g. the nominal value of a certified standard material). If a measurement is not accurate, this can sometimes be due to a systematic error. Often this can be overcome by calibrating and adjustment of instruments.
> - Precision: This describes the closeness of results of measurements of the same measured quantity under the same conditions, i.e. the same thing is measured several times. It is often quantified as the standard deviation of the values around the average. It reflects the fact that all measurements include a random error, which can be reduced, but not completely eliminated.

- Uncertainty: This term characterizes the range within which the true value is expected to lie with a specified level of confidence. It is the overarching concept which combines precision and assumed accuracy. As shown in Figure [5-1], measurements can be accurate, but imprecise, or vice versa. The ideal situation is precise and accurate."

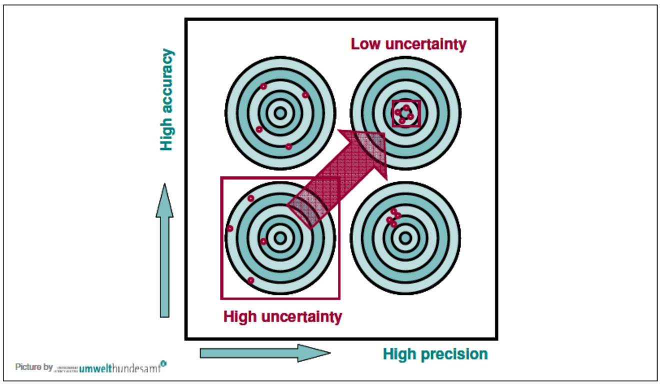


Figure 2—1 Illustration of the concepts accuracy, precision and uncertainty. The bull's eye represents the assumed true value, the "shots" represent measurement results. (EC, 2012)

The thesis analysis is predominantly based on secondary data, that is, data that the author had no control over designing and collecting. Concerns over the meta-data (data about the data) include the position, accuracy and resolution of meters and sensors. Little is known about the data collection in the European trials and whilst the confidential EST Technical Monitoring Specification (Martin & Watson, 2008) does provide a list of monitoring equipment, it states (p10): "the table is not intended to be prescriptive, the ultimate choice of equipment lies with the monitoring contractor." Importantly the Technical Monitoring Specification is just that, a specification, and not a review of the actual monitoring and data collection process. The location of sensors is critical if operational models are to be built on their output. Flow meters need to be inserted in straight pipe of a minimum length, both before and after, if accuracy is to be taken as that assigned by manufacturers. Nothing is known of the position of temperature sensors such as whether they are inside the heat pump casing or of their distance from key components.

The technical monitoring specification did not call for a full energy balance to be carried out to validate the data received from the field trial, only nine ground source and no air source heat pump sites were set up with such a provision. Gastec make the following point in the Consultation to Technical Monitoring Specification, (EST, 2008 pp6-7):

> "mCHP [micro combined heat and power] and CT [unknown, perhaps low carbon technologies] and EST condensing boiler field trials shows that it is necessary to validate data by use of an energy balance. Otherwise, wrong data can be accepted and the results

> of the whole trial are invalidated. This is not because of faulty instrumentation (although this can happen despite test certificates) but because of unpredictable interactions between the equipment and the installation…. In an ASHP this becomes difficult. Measuring the air flow across an air source heat pump is almost impossible to do with the required accuracy, and in this case, it would be necessary to measure the heat flows within the heat pump itself (ie heat flows in the working fluid of the heat pump). Whilst this is difficult and expensive because of the temperatures, pressures, and physical properties of the working fluid, it is not impossible, and *should be a requirement of the tender documents. One of the principle reasons for the success of the mCHP field trial is the unarguable quality of the data proven by the 'traffic light' test*." [author's italics].

Also of concern is the issue of missing data. Many of the EST excel files are missing data for a variety of reasons including late entry into the trials. Out of the 76 heat pumps monitored for 364 days, 26% have data for each day of the trial, 65% have data for more than 350 days, whilst 13% less than 300 days. For only 42% of the installations is the data 100% complete.

The EST confidential Technical Report (EST, 2010 p76) states in response to ANCOVA analysis:

> "The very limited success in explaining the variations in measured performance in terms of measured factors suggests that there are other influences at work."

Further complicating any statistical analysis is the sample size that reflects heat pump operation only. A taxonomical analysis, identifying both system morphology and system measurement boundary indicates the number of such systems to be 9/75. In addition, many households use unmetered living room fires that will certainly impact on measured living room temperatures. Small samples, combined with data uncertainty, suggest alternative methods may be appropriate. At this point, the analysis of clusters and single case studies becomes apparent, not simply from a statistical data approach but one that combines the content of the data files with the descriptive text in the various reports supplied to the trial sponsors. This analysis identifies a failure in consistent design practice.

The thesis then addresses design and installation requirements, focusing on the UK Microgeneration Certification Scheme in order to assess the level of technical understanding required to correctly design and install a heat pump. The analysis, using example calculations, shows that each stage of the design process is iterative and requires the designer to make decisions based on an engineering judgement that is not required for other heat sources and is not assessed in vocational education and training at typical site operative level. When comparing UK educational and technical demands for employment within the domestic heating industry with those of the European heat pump industry there appears to be a qualitative difference in training based on the concept of competence.

Research objectives

Research objectives may be summarised as:

1. What is the UK central heating *status quo* and how does it relate to heat pump policy in a UK and EU context.
2. What does theoretical thermodynamics tell us about the efficiency of heat pumps and their practical limits?
3. What is the meaning of manufacturers' COP and its relation to SPF? What and where do we measure to identify as-installed efficiency for maximum analytical and practical benefit? What are the appropriate boundaries for pragmatic outcomes?
4. What do existing heat pump trials tell us? A comparison of real installations across Europe.
5. What does a deep analysis, both quantitative and qualitative, of EST raw data tell us?
6. What are the current MCS requirements for designing heat pumps, how do they reflect on the EST trial data and what is the appropriate form of VET?

The programme of work is as follows:

- A review of heat pumps in the context of European Union and United Kingdom policy and legislation.
- A review of the relevant thermodynamic principles which govern heat pumps and their relationship to the EST trial installations, manufacturers' data and seasonal performance factor.
- A review of manufacturer test regimes and their application to seasonal performance factor. The work is supported by a pilot study on an air source heat pump producing domestic hot water at the Building Research Establishment.
- A meta-analysis of European heat pump trials based on boundary classification.
- Exploratory data analysis to develop a taxonomical approach to the EST trial methodology, heat pump and system design and pathologies, space heating and hot water loads, and occupant control. The approach to the EST data has evolved over the course of the research in conjunction with the author's role as a reviewer for the subsequent DECC technical reports of the trials (Dunbabbin et al, 2012 and 2013) and further influenced by the work of SEPEMO. What began as an investigation into field trial methodology and optimising heat pump selection through quantitative analysis of

EST data has morphed into a mixed methods approach to reflect the wide range of as-installed heat pump performance observed and the potential reasons for this.

- The cross analysis of EST heat pump design information with Microgeneration scheme installer standards and reference materials.
- From the above, the research attempts to establish the critical factors associated with trial design, measurement and vocational education and training for maximising heat pump SPF. These critical factors should also apply to the adoption of other renewable new technologies in other countries with similarly underdeveloped markets.

The EST heat pump trials of 2009 - 2010 have provided the core data for the research. All are heat pumps, all are residential, all are vapour compression, however, each case study has a different building, with variation in installation, with different occupants and installed by a range of unknown contractors. There are a range of heat pump applications (space heating only, space heating with domestic hot water) and heat pumps both with and without backup (air, exhaust, ground, water and combinations thereof), a range of manufacturers and a range of individual manufacturer's heat pump models. Most importantly, there is variation in the design of metering.

The literature review has shown that, at the time when the trial was being designed, there were no explicit performance targets, no agreed system boundary convention or any formal assessment of heat pump design and installation competence. Whilst assessment of the "technology readiness level" of heat pumps may have been the objective, it would appear that there is no simple comparison with a single 'circuitboard' designed for a specific NASA flight or MOD weapon.

Among the landscape of possible PhD methodologies for investigation, data gathering and data analysis, the research is centred around a large dataset made up of individual case studies; that is, analysis of the whole as individual cases with unique specifics, within a set of common rules derived from thermodynamics and engineering practice. The trial data is provided in 85 separate numerical excel.csv format files, accompanied by descriptive text on each dwelling and of which, 71 files provide sufficient data for in-depth exploration of heat pump performance.

The analysis of trial data is especially suited to "exploratory data analysis" (Tukey 1977, du Toit et al 1986, Myatt 2007).

> "Tukey has expounded on the practical philosophy of data analysis which minimizes prior assumptions and thus allows the data to guide the choice of appropriate models. … The variety of approaches, as well as the alternative analyses…. serves to emphasize that practical applications of data analysis generally do not lead to a 'correct' answer. The analyst's judgment and the circumstances surrounding the data also play important roles," (Velleman & Hoaglin, 2004).

The data is treated first for "displays" to visually reveal the behaviour of the data and the structure of the analyses, followed by "residuals" which focus attention on what remains of the data after some analysis. Whilst this approach provides useful graphical and tabular interpretations of performance, it also identifies the limitations of a purely statistical analysis of the full data set and leads to the development of smaller clusters of similar systems and individual case studies.

Case study is often seen as being unable to provide epistemic knowledge due to its subjective nature, "for example is not proof", to quote the Yiddish proverb. Flyvbjerg (Flyvberg, 2006, p221), in contrast, makes the following comments:

> "To understand why the conventional view of case-study research is problematic, we need to grasp the role of cases and theory in human learning. Here two points can be made. First, the case study produces the type of context dependent knowledge that research on learning shows to be necessary to allow people to develop from rule-based beginners to virtuoso experts. Second, in the study of human affairs, there appears to exist only context dependent knowledge, which, thus, presently rules out the possibility of epistemic theoretical construction."

Each heat pump installation is more than a piece of machinery governed by the laws of thermodynamics since it is the product of labour and the outcome of social structures associated therewith, and will represent the practice of labour from excellent to poor; a study of heat pump installations is therefore also unavoidably a study of social science. However, in the case of physical systems such as heating installations, thermodynamics and engineering provide an underlying theoretical construction with which to grasp the context dependent factors that arise in each case study. Case study offers, in this scenario, both context-dependent and independent knowledge. We may recognise, for example, that a profusion of incorrectly sized heat pumps tell us something about the level of design knowledge in the domestic heating industry. Flyvberg (Flyvberg, 2006, p224) quotes Eysenck (1976) who originally regarded the case study as nothing more than a method of producing anecdotes:

> "Sometimes we simply have to keep our eyes open and look carefully at individual cases - not in the hope of proving anything, but rather in the hope of learning something."

Philliber (Philliber, et al 1980) identify case study as a "blueprint" dealing with at least four problems: what questions to study, what data are relevant, what data to collect and how to analyse the results. Yin (Yin 2009 p27) suggests five critical components that may be adapted to the EST trials:

- What questions to ask, who, what, where, how, why.
- What propositions, where: "each proposition directs attention to something that should be examined within the scope of the study. [The proposition] tells you where to look for relevant evidence". The researcher must frame their data gathering within a sound theoretical and practise-based context.
- The units of analysis. This is an attempt to put a boundary around the "case", where the questions and propositions define the limit of the study. With regard to the field trials, the prime units of analysis are the Watt-hour and system temperatures which can be developed to provide quantitative outputs, but to more fully understand the context, the research must also attempt to prise out or extract as much qualitative data as is available from heat pump manufacturers' data sheets and the trial literature including descriptive text and photographs.
- Logic linking data to proposition and criteria for interpreting the findings, including: "pattern matching, explanation building, time-series analysis, logic models and cross-case synthesis".
- Criteria for interpreting a study's findings. Any data analysis must be founded on an understanding of first principles, outputs first assessed by simplified methods such as 'exploratory data analysis' (EDA) to identify macro issues such as seasonal performance followed by the micro issues that identify reasons for any particular individual level of performance, or indeed, the inability to pronounce on such reasons.

Yin warns the reader that individual case studies are not simply "sampling units" to be assessed by statistical generalisation such as probability but that the mode of generalisation is:

> "analytic generalisation, in which previously developed theory is used as a template with which to compare the empirical results of the case study. If two or more cases are shown to support the same theory, replication may be claimed. The empirical results may be considered yet more potent if two or more cases support the same theory but do not support an equally plausible, rival theory."

Importantly, it is this level of analysis that gives rise to policy implications for the energy and carbon reduction potential of heat pumps.

Yin states that four tests are common to all social science methods, tests which have been summarised in "numerous books":

- Construct validity: identifying correct operational measures for the concepts being studied
- Internal validity: seeking to establish a causal relationship where certain conditions are believed to lead to other conditions as distinguished from spurious relationships
- External validity: defining the domain to which a study's findings can be generalised
- Reliability: demonstrating that the operations of the study – such as the data collection procedures – can be repeated with the same results. (Yin 2009 p40)

An initial review of the potential for a formal case study analysis provides the following observations: The research has no control of the selection of the EST trial dwellings or of the data collection protocol and thus can only comment on the "construct validity" from an observational viewpoint. To be useful, the research must therefore rely on the researcher addressing Yin's: "Logic linking data to proposition and criteria for interpreting the findings". Some understanding of "Internal validity" is attempted by a calculation of seasonal performance factor at the various system boundaries, through STATA statistical software analysis of air and ground source heat pumps (see Appendix). Drilling deeper in order to establish internal and external validity, each case study may be initially assessed under three main categories: the interaction between the building and technology, the building occupants and the monitoring protocol, Table 2-1.

Case study	Physics	Occupant	Monitoring
1	Building heat loss Heat only or heating and hot water Heat pump type Heat pump power Type of emitters DHW load Resistance backup etc	Number of occupants Hours of occupation Continuous/intermittent heating Space heating control pattern DHW use etc	Meter positions: 1) Electricity to HP 2) Heat from HP 3) Space heating 4) Primaries 5) DHW draw off Data quality & completeness etc

Table 2—1 Initial case study evaluation

Such a matrix provides an overview to assess commonalities between different systems and an initial tool for data analysis. The matrix provides visual and numerical overviews to assess, among other things, monitoring protocols, hot water use and heat pump sizing to building heat loss ratio, in an attempt to identify the independent variables which impact on seasonal performance.

The matrix is, however, subject to the requirement that the monitoring protocol, and hence the raw data collected, is the same for all installations and whilst it presents a logical approach to Yin's "pattern matching, explanation building, time-series analysis, logic models and cross-case synthesis", in this study, it has been found that there is little consistency between individual installations with which to build up such pattern matching, an observation which became apparent upon classifying trial installation components, boundaries and metering.

Whilst recognising that each installation is discrete, the *raison d'etre* of a field trial is that wider inferences reflecting a population will be drawn from the sample. Stake (Stake 1995) promotes:

> "triangulation protocols or procedures… efforts that go beyond simple repetition of data gathering to a deliberative effort to find the validity of the data observed."

Importantly he refers to "consequential validity" where the consequences of others using the same measurements should be considered part of the researcher's responsibility. Triangulation based on Yin's "sound propositions" and "construct validity" must reflect just how crucial the study will be in providing clarity for future decision making. Heat pump efficiency must be compared to other heat sources with transparency, a common metric, and a long term view applied to the carbon reduction potential based on the future projections of carbon intensity of the electrical grid.

Myatt (Myatt 2007) describes the process of "making sense of data" as requiring a methodology based on four steps:

- Problem definition: The problem definition is clearly the comparison of the various field trials in order to pronounce on the relative efficiencies of heat pumps and their competitors.
- Data preparation: The heat pump data will require preparation to enable comparisons to be made.
- Implementation of the analysis: Myatt describes implementation of the analysis in three stages under the title of "exploratory data analysis and data mining".
    - Summarising the data
    - Finding hidden relationships

- Making predictions

- Deployment of the results: It is expected that the research will provide feedback to energy suppliers, manufacturers and the wider community.

Such an approach is visualised in Figure 2-1:

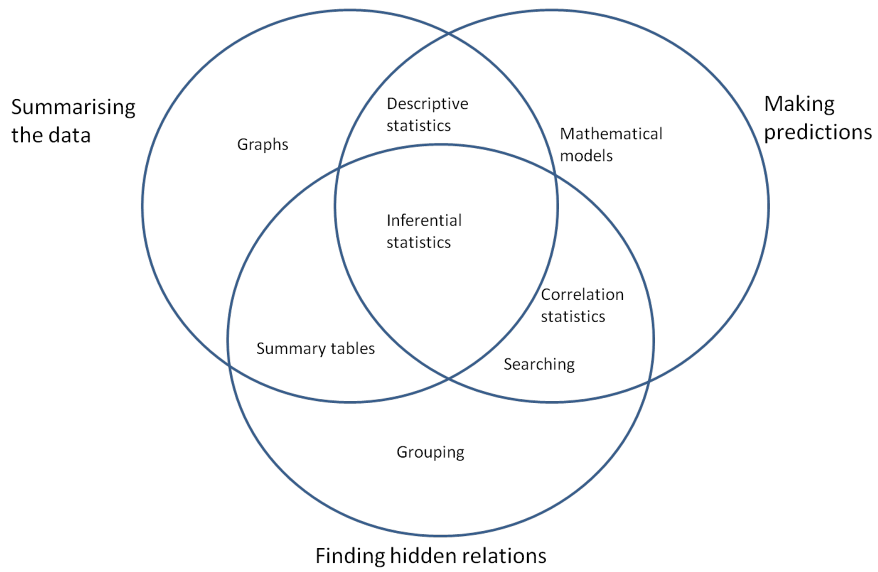


Figure 2—2 Data analysis tasks and methods (Myatt 2007)

The approach is dependent on data collection consistency since all systems need to be measured in the same way to develop the core objective - inferential statistics allied to case study observation that tells us more than is currently available.

The thesis provides a review of the relevant literature within each chapter, exploring the texts within the context of their usage. The literature review is an historical snapshot up to September 2013 within a field of study that is constantly evolving as new reports are published, legislation delivered and training documentation enhanced. The Sepemo online database[1] of European heat pump installations continues to expand providing further trial results at SPF_{H3}. The on-going analysis of heat pump performance is far from over.

[1] Sepemo Field test sites: http://www.sepemo.eu/field-test-sites/

Chapter 3 Thermodynamics of heat pumps

Introduction

We have seen in Chapter 1 that EU member states are promoting the heat pump as a low carbon technology, a source of renewable heat. This chapter aims to provide a thermodynamic analysis of heat pumps to identify the source of this renewable heat, the terminology and concepts. As well as describing practical heat pump cycles, this chapter also maps heat pump components and practicalities onto the spectrum of heat pumps in the EST field trials.

Since the primary aim of the research is to identify heat pump installation efficiency, it is essential that the underlying science of thermodynamics is understood and applied where appropriate. The chapter opens with a description of heat and work based on the first law of thermodynamics. Because heat pumps are 'reverse heat engines', the chapter describes the principles of the heat engine and the concept of the most efficient or ideal heat engine, the Carnot cycle. Real cycles cannot be 100% efficient, they are irreversible and a number of corollaries arise from the analysis including the second law of thermodynamics, the 'Clausius inequality' and entropy generation for irreversible cycles. The chapter then describes refrigeration and heat pumps, the cycle components, processes and their analysis using temperature-entropy and pressure-enthalpy diagrams to establish efficiency or coefficient of performance. There is an introduction to compressor efficiency and the role of the expansion valve as well as more innovative technologies such as desuperheating and the transcritical cycle. Finally the chapter applies real operating conditions to the ideal cycle model and draws the conclusion that a single value of COP is insufficient to describe the performance of any given heat pump system.

Thermodynamics

Heat flows from hot to cold and has the potential to generate work, whilst work done on a system has the potential to move heat from cold to hot. The observation of these processes, the development and application of the Laws that govern it is the subject of thermodynamics. Thermodynamics as a science developed, as a result of the Industrial Revolution in the late 18^{th} and 19^{th} centuries, principally through the analysis of heat engines for the extraction of useful mechanical work. From this study of heat and work arose two fundamental laws of thermodynamics; the first, that energy is conserved and the second, that an engine of 100% efficiency is not possible. Heat and work are both forms of energy and have the same units, the Joule. The process of heat transfer across a temperature difference is irreversible. Similarly, the resistance in an electrical circuit generates heat but transferring this heat to a circuit will not generate an equivalent amount of electricity, which is a form of work. Consider a paddle wheel immersed in water with a frictionless axel attached to a weight. Dropping the weight will generate heat in the water, James Prescott Joule's famous experiment, but no amount of heat in the water will cause the paddle wheel to rotate. Work is done on the system and heat is generated, the process is irreversible. Thus although a sum of heat and work may both contain the same amount of energy in Joules, they are not equivalents. High temperature heat energy has a greater potential to create work than low temperature energy, hence heat energy has 'quality' and 'work' has the ultimate 'quality'.

Processes operate as either closed or open systems. A closed system has a "control volume" with a moving boundary where heat (Q) and work (W) may cross the boundary affecting the internal energy (U), the pressure and volume. Heat into the system is positive, heat out negative, work-in negative and work out positive. If we assume that a system is designed to produce work from heat then the change in the energy of the system after a process is:

$$\Delta \boldsymbol{E} = \boldsymbol{Q}_{in} - \boldsymbol{W}_{out}, \qquad \text{Equation 3-1}$$

where the total energy (E) in Joules has three components expressible in SI unit dimensional analysis as:

1) internal energy of the molecules (U): Nm, = (kg x m/s^2) x m = M x LT^{-2} x L = ML^{2}T^{-2} = (J),
2) the kinetic energy (KE): $m\left(\frac{V_2^2}{2} - \frac{V_1^2}{2}\right)$, = M x (LT^{-1})^2 = ML^{2}T^{-2} = (J)
3) and the potential energy (PE): $mg(z_2 - z_1)$, = M x LT^{-2} x L = ML^{2}T^{-2} = (J)

In most systems KE and PE have a minor impact and can be ignored leaving Q – W = dU.

A steady flow open system such as a cycle, consist of a series of control volumes. A control volume may be modelled as a closed system with boundary work to move the mass flow across the control boundary. Since work equals force (F) times distance moved (ds), or W = F.ds, and force = pressure x area, W = PAds, and Ads = V, therefore boundary work = PdV. The energy change for a control volume can therefore be expressed as Q – W = U + PdV, where u + Pdv = h, the system enthalpy.

Thermodynamic processes may be described as reversible and irreversible. A reversible process is defined as one that can be reversed without leaving any trace on the system and surroundings but may be better understood by considering irreversibilities – the real world. Relevant irreversibilities include friction, unrestrained expansion and heat transfer across a finite temperature difference. All real systems generate friction between moving parts and resulting from fluid flow. Friction occurs when two bodies in contact move relative to each other, such as a piston in a cylinder; work is needed to overcome this friction and the work is converted to heat. As the piston returns to its original position, the heat is not converted back to work but further heat is generated. The process is clearly not reversible. Reversible expansion and contraction occur in "quasi-equilibrium" where the system contents, the gas:

> "remains infinitesimally close to a state of equilibrium at all times….[where upon compression] if the piston velocity is not very high, the pressure and temperature will increase uniformly throughout the gas. Since the system is always maintained at a state close to equilibrium, this is a quasi-equilibrium process" (Cengel & Boles, 1994 p258).

Piston compression and expansion generate work as pressure times change in volume ($\frac{N}{m^2} \times m^3 = Nm = \ Joules$). Rapid compression causes both turbulence and the fluid to build up at the piston face, demanding greater work-input, whereas rapid expansion reduces the pressure at the piston face due to non-equilibrium distribution of the working fluid. In practice more energy is needed for compression than is achieved in expansion therefore the process is irreversible. Real heat engines must operate with finite temperature differences across the boundary between the system and its reservoirs since heat flows from hot to cold. The only way to reverse heat flow, from cold to hot, is by refrigeration, which will require work and therefore is, in practice, irreversible; it follows that real systems are irreversible. However, the concept of reversibility is useful since, in the same way that "ideal fluids" provide an initial approach to investigate fluid dynamics, reversible heat engines provide the means to identify the "ideal" heat engine and the fundamentals of their operation. Such simply defined limiting states are always a useful guide to thought.

Heat engines

Heat may be harnessed to produce work-in a thermodynamic cycle known as a heat engine, the basic components are shown in Figure 3-1.

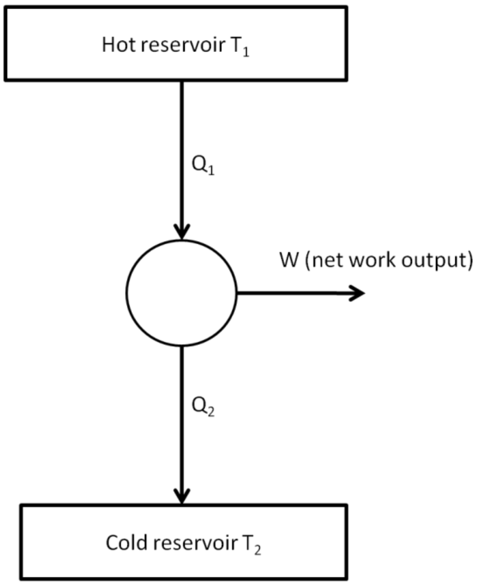


Figure 3—1 The heat engine

Heat (Q_1) from a high temperature reservoir (T_1) is used to drive an engine and work is produced (W_{out}). For the heat engine to continuously produce work, that is to operate in a cycle, heat (Q_2) must be transferred to the low temperature reservoir (T_2). This need for a low temperature reservoir is neatly described by Cengel and Boles (1994, p246) using the apparatus shown in Figure 3-2.

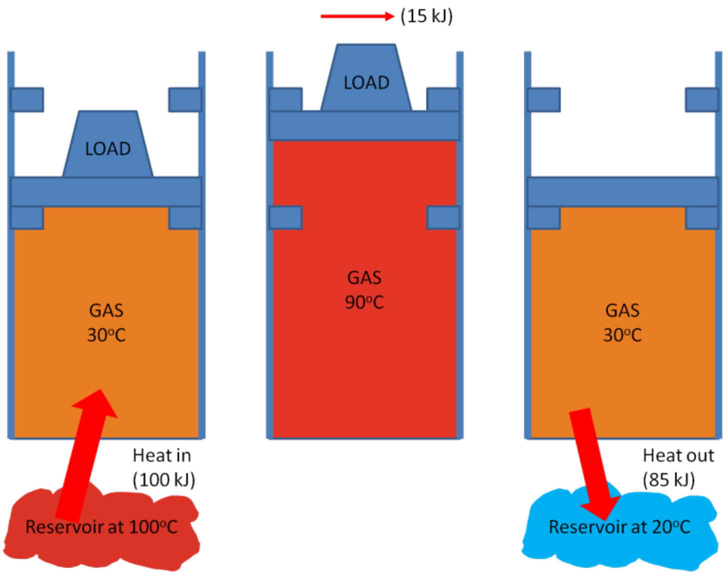


Figure 3—2 A heat engine cycle cannot be completed without rejecting some heat to a low temperature sink (Cengel & Boles, 1994)

To paraphrase: the heat engine is designed to lift weights between the two stops on the piston-cylinder device using the gas contained in the cylinder as the working fluid. 100 kJ of heat from the high temperature reservoir at 100°C causes the gas to rise in temperature and expand, lift the weight and produce say 15 kJ of work by increasing the potential energy of the system. The weight is removed, the gas temperature is say 90°C. It is not possible to transfer the 85 kJ back to the reservoir at 100°C for later use since the temperature of the piston gas is less than that of the reservoir. We must transfer heat to a low temperature reservoir to drop the temperature back to 30°C and complete the cycle. We conclude that it is not possible to have a 100% efficient heat engine and in this particular case, the efficiency is as little as 15%. Figure 3-2 also illustrates the Kelvin-Planck statement of the second law:

> "It is impossible for any device that operates on a cycle to receive heat from a single reservoir and produce a net amount of work" (Cengel & Boles, 1994 p248).

A cyclical heat engine must have two reservoirs.

The efficiency of a heat engine is best described as the ratio of what is desired divided by what is required; efficiency of a heat engine = desired /required, or $\eta_{th} = \frac{W_{out}}{Q_H}$.

Since $\boldsymbol{W_{out} = Q_H - Q_L}$ **then,** $\boldsymbol{\eta_{th} = \frac{Q_H - Q_L}{Q_H}}$ Equation 3-2

or, $\boldsymbol{\eta_{th} = 1 - \frac{Q_L}{Q_H}}$ Equation 3-3

Since efficiency is dependent on a ratio of Q_H and Q_L, the efficiency must be dependent on some property of the heat reservoirs, the source of Q_H and Q_L; Thompson (Lord Kelvin) proposed that it was their temperatures. Whalley (1992 p95) and Cengal & Boles (1994 p267) both describe Thompson's thought process using three reverse heat engines operating between hot, cold and intermediate temperature reservoirs, Figure 3-3.

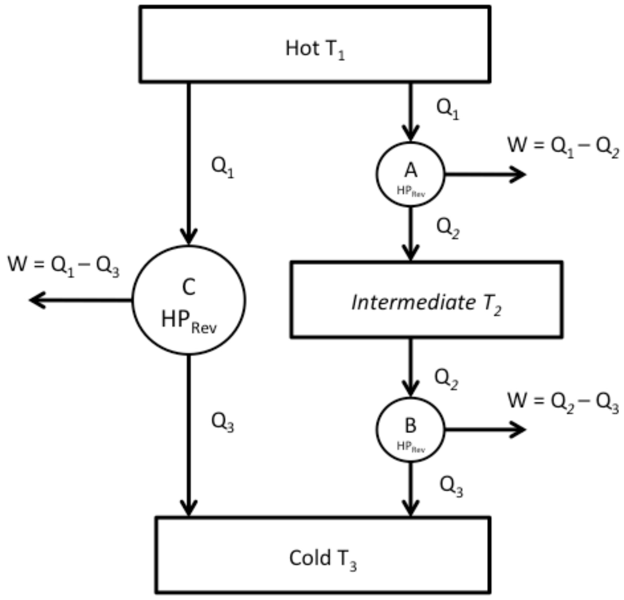


Figure 3—3 Three reversible heat engines operating between hot, cold and intermediate temperature reservoirs (after Cengal, 1994)

Thompson argued that if $\eta_{th} = 1 - \frac{Q_L}{Q_H}$, then $\frac{Q_L}{Q_H}$ could be a function of the reservoir temperatures, thus $\frac{Q_H}{Q_L} = f(T_H, T_L)$

where $Q_H = T_1 \; and \; Q_L = T_3$, the reservoir high and low temperatures.

The amount of heat rejected by engines B and C must be the same, since A and B can be combined into one reversible engine operating between the same reservoirs as C and thus A and B combined have the same efficiency as C.

These heat transfers can be expressed as ratios and some form of temperature function:

$$\frac{Q_1}{Q_2} = f(T_1, T_2)$$

$$\frac{Q_2}{Q_3} = f(T_2, T_3)$$

$$\frac{Q_1}{Q_3} = f(T_1, T_3)$$

Since $\frac{Q_1}{Q_3} = \frac{Q_1}{Q_2} \times \frac{Q_2}{Q_3}$ we can also write:

$$f(T_1, T_3) = f(T_1, T_2) \times f(T_2, T_3)$$

Cengal & Boles state:

> "A careful examination of this equation reveals that the left hand side is a function of T_1 and T_3 and therefore the right hand side must also be a function of T_1 and T_3 only and not T_2. That is the value of the product on the right hand side is independent of the value of T_2. This condition will only be satisfied if the function *f* has the following form:
>
> $f(T_1, T_2) = \frac{\phi(T_1)}{\phi(T_2)} \; and \; f(T_2, T_3) = \frac{\phi(T_2)}{\phi(T_3)}$ so that $\phi(T_2)$ will cancel yielding:
>
> $$\boldsymbol{\frac{Q_1}{Q_3} = f(T_1, T_3) = \frac{\phi(T_1)}{\phi(T_3)}} \qquad \textbf{Equation 3-4}$$
>
> For a reversible heat engine operating between two reservoirs at temperatures T_H and T_L, Equation 3-4 can be written as:
>
> $$\frac{Q_H}{Q_L} = \frac{\phi(T_1)}{\phi(T_3)}$$
>
> This is the only requirement that the second law places on the ratio of heat flows to and from the reversible heat engines. Several functions will satisfy this equation and the choice is completely arbitrary. Thompson first proposed taking $\phi(T) = T$ to define a thermodynamic temperature scale." (Cengal & Boles, 1994 pp267-268)

Rogers and Mayhew take Thompson's definition of reversible heat engine efficiency and show how a linear thermodynamic scale can be derived from Equation 3-4 in the form of $T = T_H \frac{Q_L}{Q_H}$ where all temperatures are greater than zero (Rogers & Mayhew, 1992 pp63-66). The thermodynamic scale of temperature, where absolute zero is 0 Kelvin, demands that all temperatures have positive values. We conclude that a heat engine converts heat into work:

$$Q_1 - Q_2 = W, thus\ Q_1 - Q_2 - W = 0,\ Q_1 > W, \therefore\ Q_2\ must\ be > 0\ .$$

No heat engine operating in a cycle can be 100% efficient; this is an expression of the second law of thermodynamics and provides the basis for exergy or availability analysis of cycles.

The Carnot cycle

The question arises, what is the highest efficiency that can be obtained? If we imagine a frictionless cycle where compression and expansion are in quasi-equilibrium and where heat transfer is effectively isentropic as the temperature difference between the heat reservoirs and the engine approaches zero. Such a cycle was imagined by Sadi Carnot[2], the Carnot cycle, consisting of four processes and providing the upper limit to second law efficiency, Figure 3-4.

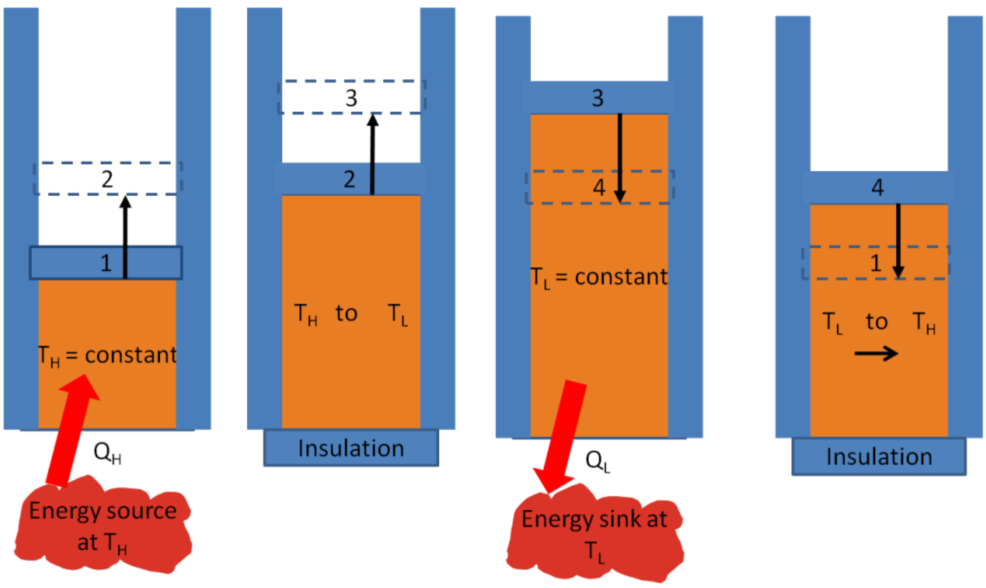


Figure 3—4 Carnot cycle in a closed system (Cengel & Boles, 1994 p262)

[2] Carnot S, N, L. (1824) *Réflexions sur la puissance motrice du feu et sur les machine propres à developer cette puissance.* Reflections on the motive power of fire and on machines fitted to develop that power. Bachelier

1-2) The system is placed in contact with a high temperature reservoir. Heat from the high temperature reservoir into the system across a differential temperature difference increases the energy of the system causing isothermal expansion; the piston does work on the surroundings.
2-3) The system is insulated. The higher temperature and thus energy of the system continues to work on the surroundings until the expansion of the gas causes the system temperature to drop until it is in equilibrium with that of the surrounds. This is adiabatic expansion.
3-4) The insulation is removed and the system is placed in contact with a cold reservoir. The differential temperature difference causes the gas to contract; the surroundings do work on the system maintaining isothermal compression.
4-1) The system is insulated. The surroundings do work on the system causing the temperature to rise, adiabatic compression, until the temperatures are once more in equilibrium.
The process may be described on a pressure-volume diagram, see Figure 3-5.

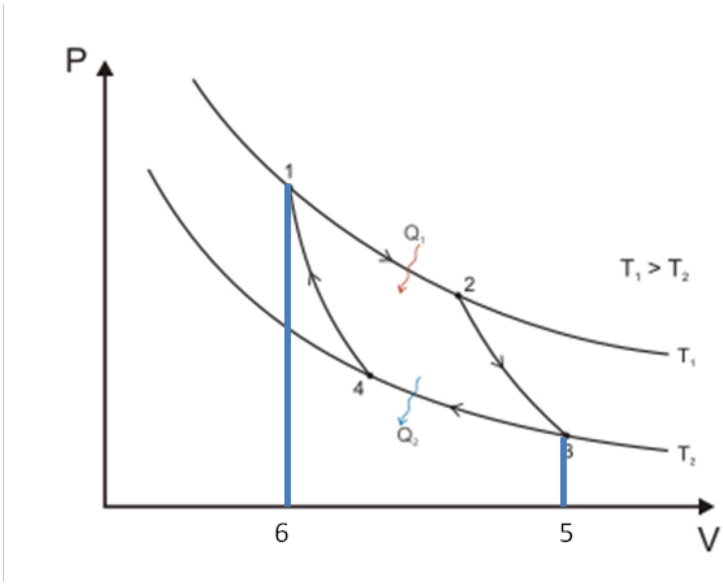


Figure 3—5 A Carnot cycle acting as a heat engine, plotted on a pressure-volume diagram to illustrate the work done (Wikimedia).

The area inside the cycle in the PV diagram is the work achieved. The net work of the system is area 1, 2, 3, 5, 6 minus area 1, 4, 3, 5, 6. Note that work is a "path" function; without knowing the path taken during expansion or contraction, it is not possible to integrate the equation for each process and calculate the net work output.

> "The Carnot cycle can also be executed in a steady flow system. Being reversible the Carnot cycle is the most efficient cycle operating between two specified temperature limits. Even though the Carnot cycle cannot be achieved in reality, the efficiency of actual cycles can be improved by attempting to approximate the Carnot cycle more closely." (Cengel & Boles, 1994 p264).

Rogers & Mayhew (1992 p66) identify:

"the most important characteristics of engines operating between only two reservoirs".

a) Reversible engines have an efficiency of $(T_1 - T_2)/T_1$
b) For a given value of T_2, the efficiency increases with increase of T_1. T_2 is generally ambient temperature of air or water so T_1 should be as high as possible for maximum efficiency.
c) For reversible engines operating between only two reservoirs, $Q_1/T_1 = Q_2/T_2$. If an engine is irreversible, its efficiency is less than $(T_1 - T_2)/T_1$. The heat rejected must be greater than that rejected by a reversible heat engine, $Q_1/T_1 < Q_2/T_2$.

Rogers and Mayhew comment:

> "It is important to know that when applying statements (a), (b) and (c) to reversible engines, the temperature T refers both to the temperature of the reservoir and to the temperature of the fluid exchanging heat with it. *On the other hand, when considering irreversible engines, T refers to the temperature of the reservoir only; the temperature of the fluid may then not be the same as that of the reservoir and different parts of the fluid may have different temperatures."* [author's italics]

This is a *critical statement* since it expresses the maximum efficiency possible from an irreversible cycle, the Carnot efficiency for real systems. We see that $(T_1 - T_2)/T_1$ may always be used to assess the maximum potential efficiency of heat engine cycles. The statement equally applies to heat pumps, reverse heat engines, where we know the source (air, ground or water) and sink (heat emitter) temperatures irrespective of the temperatures of the system fluid.

Entropy and the Clausius Inequality

The impact of system irreversibility may be quantified through the property of entropy defined by the 'Clausius inequality'. The Clausius inequality is expressed as: "Whenever a system undergoes a cycle, [the cyclical integral] $\oint(\frac{dQ}{T})$ is zero if the cycle is reversible and negative if irreversible, i.e. in general $\oint(\frac{dQ}{T}) \leq 0$" (Rogers & Mayhew, 1992 p67).

Cengel and Boles describe the property (dQ/T) thus: Consider a "System" connected to a thermal energy reservoir T_R through a reversible engine, Figure 3-6. The reversible engine receives heat δQ_R and supplies heat δQ to the System whose absolute boundary temperature is T and produces work δW_{sys}.

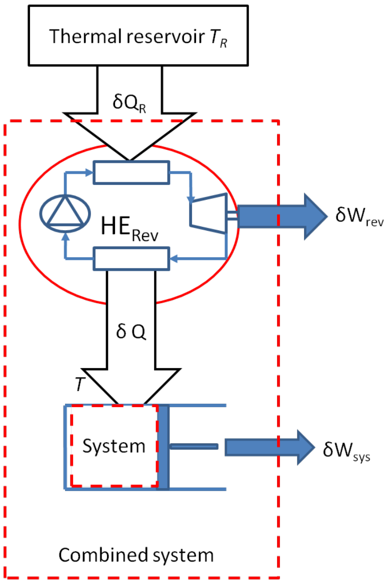


Figure 3—6 The system considered in the development of the Clausius inequality (Cengel & Boles, 1994)

Apply the conservation of energy principles to the combined system where E is energy.

$dE_c = \delta Q_R - \delta W_c, where\ \delta W_c\ is\ the\ combined\ system\ work\ output$

or $\delta W_c = \delta Q_R - dE_c$

[Consider Rogers and Mayhew point (c): for the reversible engine, $\frac{\delta Q_R}{T_R} = \frac{\delta Q}{T}$, or, $\delta Q_R = T_R\frac{\delta Q}{T}$]

Thus $\delta W_c = T_R\frac{\delta Q}{T} - dE_c$

> "Let the System undergo a cycle while the cyclic device [the reversible engine] undergoes an integral number of cycles. Then the relation above becomes $W_c = \oint T_R\frac{\delta Q}{T} - 0$ [since during a reversible cycle, the energy of a system, the net change, is zero]. It appears that the combined system is exchanging heat [$W_C = \oint T_R\frac{\delta Q}{T} = \oint \delta Q_R$] with a single thermal energy reservoir while involving (producing or consuming) work W_C during a cycle. On the basis of the Kelvin Planck statement [a heat engine must have two reservoirs]….. we reason that W_C cannot be work output and thus it cannot be a positive quantity. Considering that T_R is an absolute temperature and thus a positive quantity, we must have $\oint\frac{\delta Q}{T} \leq 0$ which is the Clausius inequality. This inequality is valid for all thermodynamic cycles." (Cengel & Boles, 1994 p297).

For internally reversible systems, where for example quasi-equilibrium compression and expansion take place: $\oint\left(\frac{\delta Q}{T}\right)_{int\ rev} = 0$.

For irreversible systems: $\oint\boldsymbol{\frac{\delta Q}{T} < 0}$ **Equation 3-5** or, $\boldsymbol{S_{gen,cycle}} = -\oint \frac{\boldsymbol{\delta Q}}{\boldsymbol{T}}$ Equation 3-6

where $S_{gen,\,cycle}$ is the:

> "entropy generation associated with a cycle, which is a measure of the irreversibilities or imperfections which occur during the cycle…..Entropy generation can never be negative" (Cengel & Boles, 1994 p297).

Entropy has units of J/K. Cengel and Boles provide a simple example to illustrate both (a) Clausius inequality and (b), to show whether the cycle violates the 2nd Law Carnot efficiency (Cengel & Boles, 1994 p298):
A high temperature source of 1000 K transfers 600 kJ into a heat engine which converts 150 kJ to work and rejects 450 kJ to a low temperature reservoir at 300 K.

(a) $S_{gen,cycle} = -\oint \frac{\delta Q}{T} = \frac{Q_H}{T_H} - \frac{Q_L}{T_L} = \frac{600}{1000} - \frac{450}{300} = -0.9\ kJ/K$

Just as heat lost from a system is assigned a negative value, so is entropy. Entropy is rejected from the system to the "universe". For any real cycle, the entropy of the universe must always increase.

(b) $\eta_{th} = 1 - \frac{Q_2}{Q_1} = 1 - \frac{450}{600} = 0.25$ $\eta_{th\ Carnot} = 1 - \frac{T_L}{T_H} = 1 - \frac{300}{1000} = 0.7$

We see that the cycle efficiency is, as it should be, less than the Carnot efficiency between the reservoirs. Where a process is adiabatic and where there are no irreversibilities within the system, it is internally reversible, the system is said to be isentropic.

This section has identified the principal thermodynamic issues associated with heat engine analysis including 2nd law efficiency, the development of the absolute temperature scale and the generation of entropy. We may now apply these functions of heat engines to 'reverse heat engines' where, instead of a high temperature reservoir exchanging heat with a low temperature reservoir and generating work, the cycle is reversed and low temperature heat is raised to a higher temperature by work being done on the system. System efficiency remains dependent on the 2nd law, absolute reservoir temperatures and, of course, real-world irreversible reverse heat engines generate entropy leading to sub-optimal Carnot efficiencies.

Reverse Heat Engines, the refrigerator and heat pump

To move heat from a cold reservoir to a hot reservoir requires work-input, this is the reverse heat engine shown in Figure 3-7. Reverse heat engines exhibit the same thermodynamic properties as previously described; they may be reversible or irreversible, the Carnot efficiency applies as the cycle is reversed. Reverse heat engines may be applied to a cooling process, where they are described as refrigerators, or to a heating process, where they are known as heat pumps.

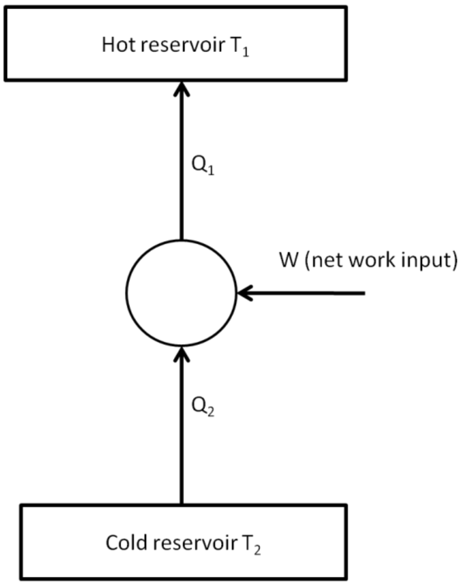


Figure 3—7 Reverse heat engine

Refrigerators and heat pumps are identical in their cycle and component parts, their only difference is in their function – what is desired of the system - and their efficiency can be expressed as desired/required.

A refrigerator is designed to remove heat from a low temperature reservoir and to dump it into a higher temperature reservoir (e.g. from freezer air temperature to kitchen air temperature). A heat pump is designed to move heat from a low temperature source, to raise its temperature above ambient (the high temperature reservoir) and to use it as a heat source for a heating system. The efficiency of refrigerators and heat pumps is described as their coefficient of performance (COP). The COP of a refrigerator is:

$$COP_{Ref} = \frac{desired}{required} = \frac{Q_L}{W_{in}} \text{ where } W_{in} = Q_H - Q_L$$

$$\boldsymbol{COP_{Ref} = \frac{Q_L}{Q_H - Q_L} = \frac{1}{\frac{Q_H}{Q_L} - 1} \text{ and } COP_{Ref\ Rev} = \frac{1}{\frac{T_H}{T_L} - 1}} \qquad \text{Equation 3-7}$$

The efficiency of a heat pump is:

$$\boldsymbol{COP_{HP} = \frac{desired}{required} = \frac{Q_H}{W_{in}}} \qquad \text{Equation 3-8}$$

where $W_{in} = Q_H - Q_L$

$$\boldsymbol{COP_{HP} = \frac{Q_H}{Q_H - Q_L} = \frac{1}{1 - \frac{Q_L}{Q_H}}} \text{ and } \boldsymbol{COP_{HP\,Rev} = \frac{T_H}{T_H - T_L}}$$ Equation 3-9

$$\boldsymbol{COP_{HP} = \frac{1}{1 - \frac{T_L}{T_H}}}$$ Equation 3-10

For the same operating conditions, heat pump COP is greater than refrigerator COP by unity (1). It is also apparent that the ratio of heat over work is greater than 1 and therefore the use of the term efficiency is inappropriate, hence coefficient of performance. One could also note that the ratio is of heat to work and therefore, whilst both are energy, they are not the same since work is organised heat, it has the ultimate "quality". Efficacy would therefore also be appropriate.

Reverse Carnot cycle

Let us imagine a reverse Carnot cycle, Figure 3-8, where the refrigerant working fluid is R12. R12 is chosen simply because its thermodynamic properties and property diagrams are widely available; R12 is a CFC refrigerant and therefore no longer manufactured for commercial use.

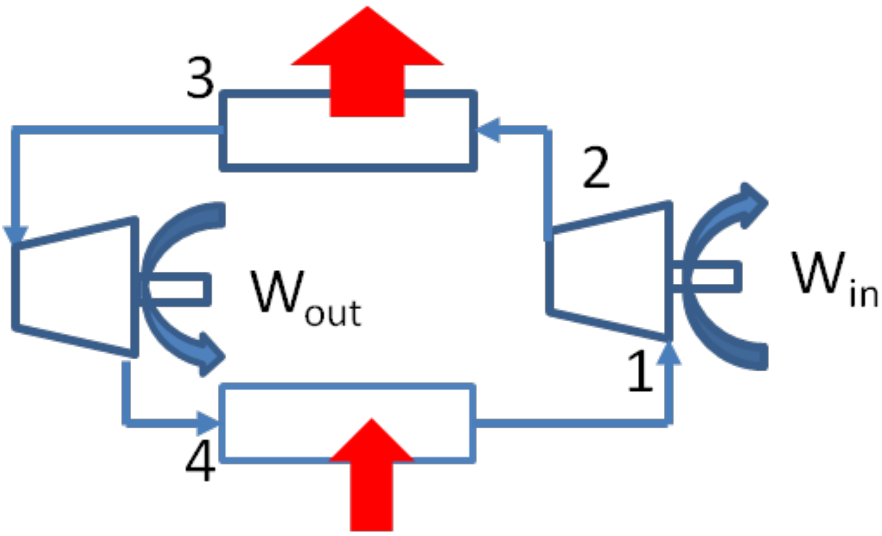


Figure 3—8 Reverse Carnot cycle

The reverse Carnot cycle can be shown in a temperature-entropy diagram (Ts diagram), produced using Coolpack software[3], Figure 3-9. The Ts diagram is represented by temperature (°C) and specific entropy (kJ/kgK) on the vertical and horizontal axes respectively. The bell or dome represents the three phases of a fluid, saturated liquid on the left and saturated vapour or gas on the right with the vapourisation process between. At the apex of the dome is the critical point, for R12 it occurs at 112°C and 41.15 bar, above which there is no clear distinction

[3] Coolpack software. http://en.ipu.dk/Indhold/refrigeration-and-energy-technology/coolpack.aspx

between liquid and vapour. Constant pressure is represented by the red lines, specific enthalpy (kJ/kg) by the blue and specific volume (m^3/kg) by the green.

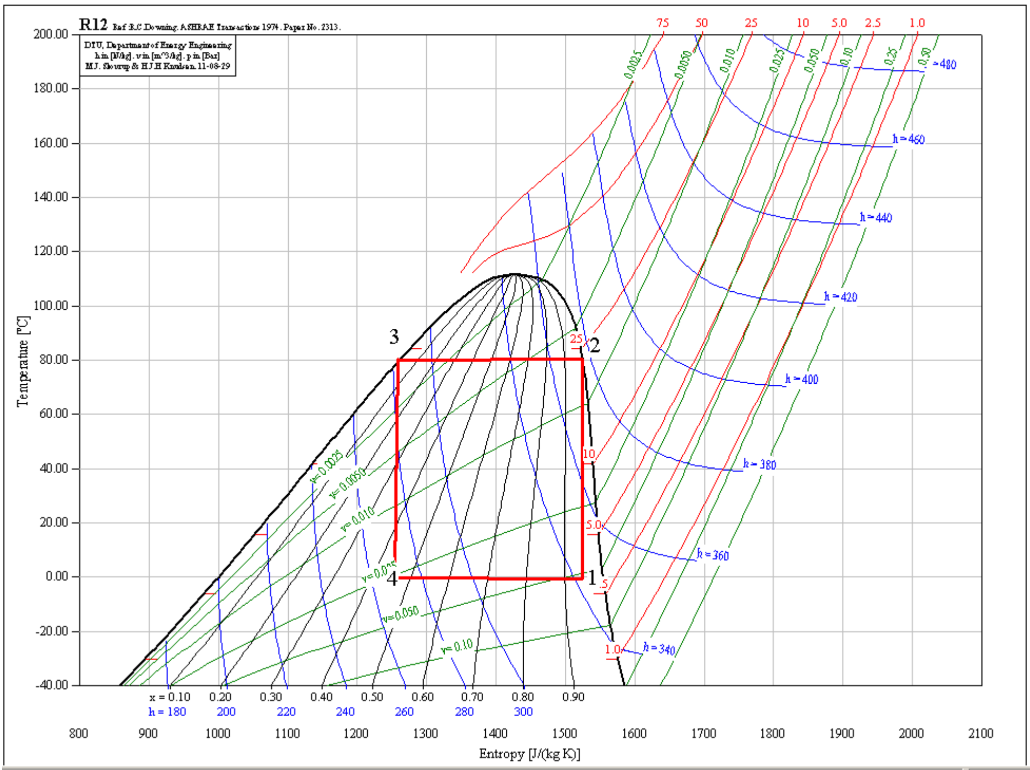


Figure 3—9 Reverse Carnot cycle, R12 Ts diagram

The reverse Carnot cycle is based on sub-critical condensation (Q_{out}) in the condenser at constant temperature, thus it is isothermal and shown as process (2-3). Isothermal evaporation (Q_{in}) occurs at the evaporator in process (4-1). Adiabatic, isentropic compression occurs at process (1-4) and isentropic expansion at process (3-4), where Figure 3-8 showed a turbine to capture this work potential. Net work output per kg of refrigerant is the area enclosed by (1-4) since T x s = J/kg. This, as we have repeatedly said, is the ideal and most efficient cycle – but it is impractical and must be compared to Figure 3-10.

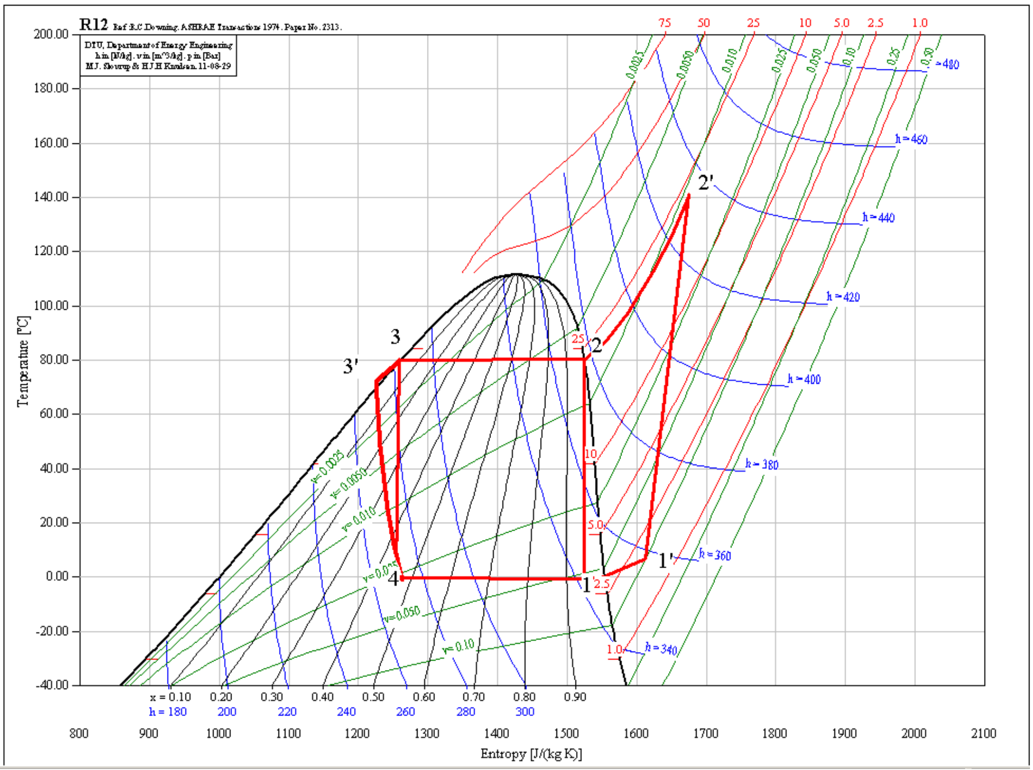


Figure 3—10 Practical heat pump cycle imposed on ideal Carnot cycle

Consider compression (1-2): a compressor cannot compress a liquid; compressors are designed to compress gases and as is evident from the Ts diagram, the process of evaporation is incomplete at point (1), the liquid-vapour mix has a "quality" of approximately 0.95 rather than 1, therefore it is not a saturated gas. Practical cycles need to ensure that the refrigerant is completely evaporated before entering the compressor and therefore rely on some "super-heat" to the gas; the process is no longer isothermal and non-reversible entropy is generated as point (1) moves to (1'). In addition, whilst isentropic (adiabatic) compression is an ideal, real compression cannot be 100% efficient due to friction and heat transfer. Friction will cause an increase in entropy whilst heat transfer from the compressor will cause a decrease. Since heat transfer is generally small in comparison with friction, compression also generates non-reversible entropy and the entropy at (1') increases to that at (2').

The process of heat transfer out of the system is both sensible (2' to 2) and latent (2 to 3). Super-heated gas exits the compressor, is sensibly cooled to the saturated gas condition (2) and then the condensation process occurs (2-3). Practical engines have some degree of "sub-cooling" (3') where saturated liquid is sensibly cooled to ensure maximum heat out at the condenser and that liquid enters the next stage. Heat exchangers increase in efficiency with size, the increase in surface area for heat transfer (u value x area). Size of heat exchanger will also impact on manufacturers' costs and on machine foot-print – the size of the unit. The handling of these three conflicting issues requires “value management” or “value engineering”, where: “Value Management is concerned with improving and sustaining a desirable balance between the wants and needs of stakeholders and the resources needed to satisfy them.” (IVM, online).

Whilst Carnot expansion provides the opportunity to extract work from the expansion part of the cycle, partly offsetting the work input in the compression part of the cycle, practical heat pumps replace the turbine with an expansion device, also known as a throttle. An expansion valve is a flow-restricting device that may be best understood through the application of the Bernoulli theorem already alluded to in describing control volumes. In simple terms, the total pressure in a system is the combination of static or “bursting pressure”, kinetic pressure and that provided by head difference – potential pressure. A flow restriction such as a capillary tube causes an increase in velocity (continuity of flow equation ($m^3/s = m^2 \times m/s$)) whereby static pressure is converted to velocity pressure leading to a static pressure drop. This pressure drop is usually accompanied by a large drop in temperature, assumed to be adiabatic ($q = 0$) and is work free ($w = 0$). Whilst expansion is isenthalpic and follows the constant enthalpy curve, it cannot be isentropic (3’ - 4) due to friction and because expansion is not in quasi-equilibrium, Figure 3-10.

The Ts diagram provides a visual image of system irreversibility but does not allow us to immediately view the energy flows into and out of the cycle based on the most obvious criteria of temperature and pressure, which are both readily measurable. Its practical application to heat engine efficiency is circumscribed by the lack of an entropy meter. For monitored systems the useful work or heat output is more easily found by direct measurement of cycle temperatures and pressures. The pressure-enthalpy (Ph) diagram is a far more practical solution to system analysis and, unlike Ts diagrams, Ph diagrams are provided by manufacturers for all commercially available refrigerants.

Pressure-enthalpy diagrams

The Ph diagram is wholly suited to analyse the heat pump cycle since compressor and evaporator temperatures are readily measurable. The cycle must extract and supply heat (sensible and latent) – enthalpy - at particular temperatures whilst the compressor does work on the system to achieve these temperatures, Figure 3-11.

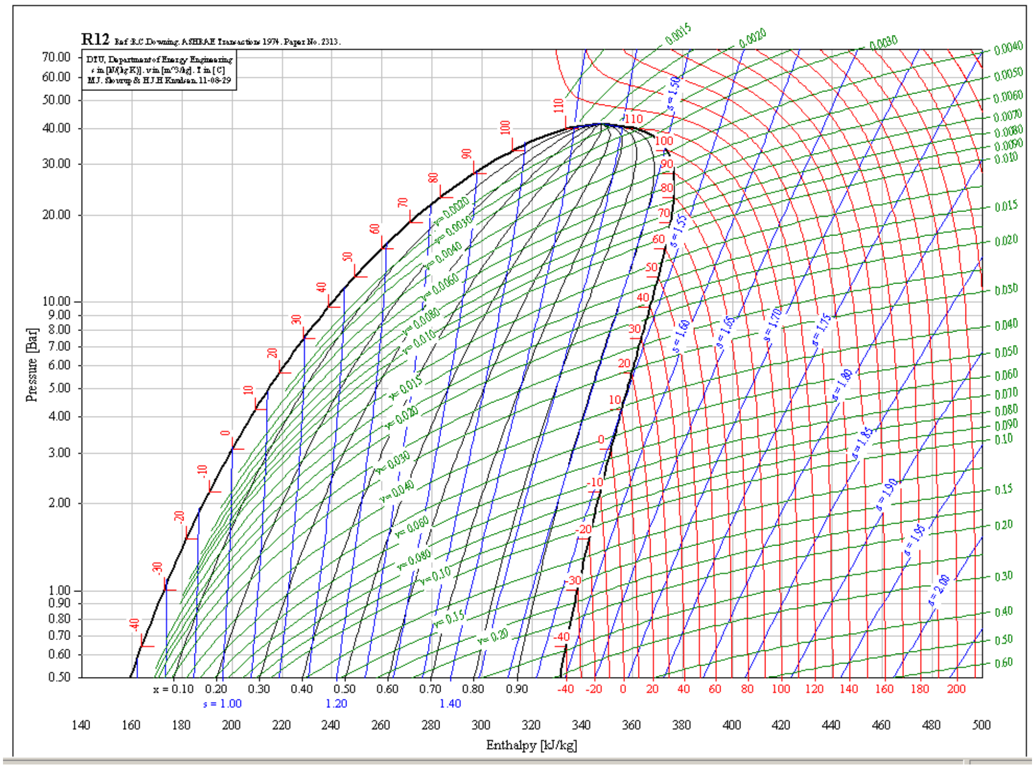


Figure 3—11 Pressure enthalpy diagram for R12 (Coolpack)

Figure 3-11 indicates the three phases of the refrigerant with a dome or bell shape indicating saturated liquid to the left, saturated gas to the right and the refrigerant critical temperature at the apex of the bell. The critical point represents the upper limit to heat transfer based on evaporation and condensation, above which there is no distinction between liquid and vapour and where heat transfer is termed "transcritical". The diagram shows pressure and enthalpy on the vertical and horizontal axes respectively. The red lines show constant temperature, the blue constant entropy and the green constant specific volume (m^3/kg). Within the dome, the black lines show refrigerant "quality", that is the percentage change from liquid to vapour. The information on the diagram is also produced in tabular format, known as the "steam tables" where enthalpy of saturated liquid is described as "hf", saturated gas as "hg" and the enthalpy of vapourisation as "hfg"; where the internationally recognised subscripts 'f' and 'g' are from the German *flüssig* (liquid) and *gesättigt* (saturated).

The heat pump cycle may be explored by applying the same conditions as used in the Ts diagram, Figure 3-10, to the Ph diagram, Figure 3-12. Evaporation occurs at 0°C and condensation at 80°C, the temperatures of the respective cold and hot reservoirs. A reversible cycle based on these conditions has a Carnot efficiency of: $COP_{HP\ Rev} = \frac{1}{1-\frac{T_L}{T_H}} = \frac{1}{1-\frac{273}{353}} = 4.41$

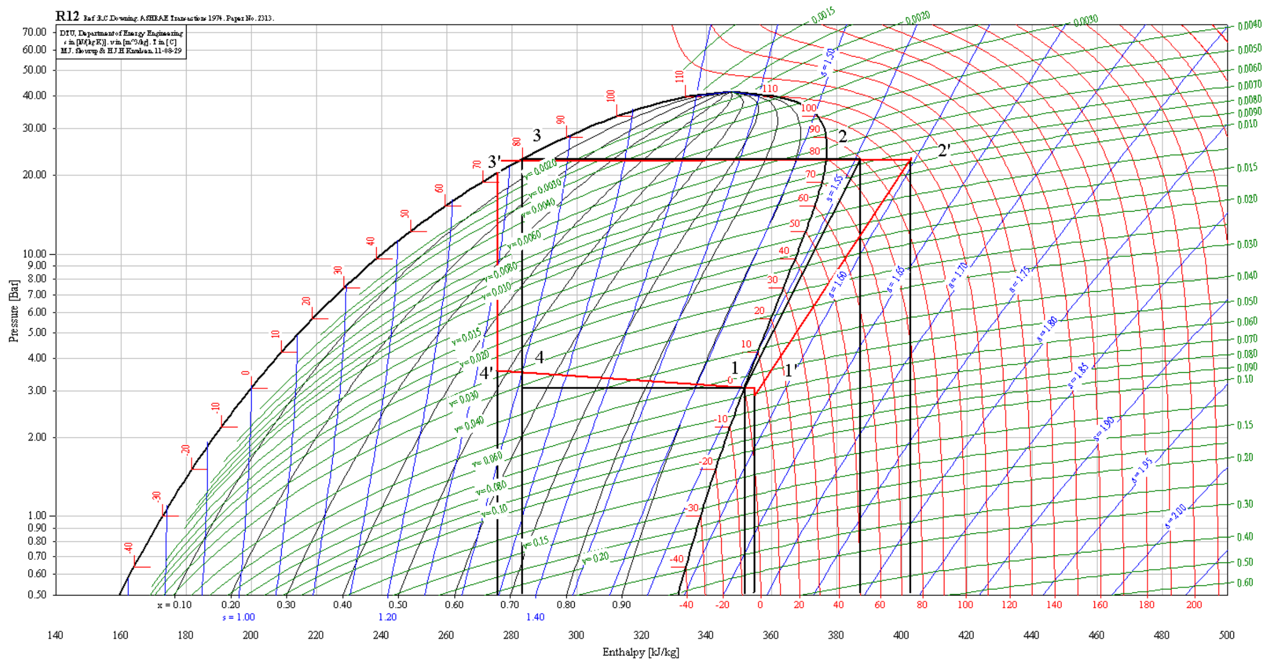


Figure 3—12 Pressure Enthalpy (Ph) diagram R12

Process 1, 2, 3 and 4 (the black lines in Figure 3-12), show isentropic compression and isenthalpic expansion, where the Ph diagram can simply be used to provide heat in, heat out and work-in (Q_{in} and Q_{out} and W_{in}).

The heat out process 2 to 3 is: (388 – 283) = 105 kJ/kg.

The Work-in process 1 to 2: (388 – 352) = 36 kJ/kg.

The resulting COP_{HP} = 2.92; the cycle is clearly not reversible.

However, a more realistic cycle (shown with the red lines) will include superheat, 1 to 1' (shown as 5K), the compressor will not be isentropic, 1' to 2', and heat out will be sub-cooled, 3 to 3' (shown as 5K); the total cycle is from points 1' to 4'. In addition, there will be pressure and temperature losses in the pipelines and heat exchangers.

Process 2'-3' is now (403 – 275) = 128 kJ/kg

Process 1'-2' is increased to (403 – 355) = 48 kJ/kg

COP_{HP} = 2.67

The additional heat transfer has increased the entropy of the cycle, lowering the COP. Clearly this is just an estimate since approximate values have been applied to system pressure drop due to friction and no allowance is made for heat gains and losses from the cycle components, or 'parasitic losses'.

The adiabatic or 'isentropic efficiency' of the compressor is:

$$\boldsymbol{\eta_C = \frac{isentropic\ compressor\ work}{actual\ compressor\ work}}$$ **Equation 3-11**

In the example, the efficiency is: (388 – 352)/(403 – 355) = 36/48 = 0.75. Non-isentropic compression demands more work input. Super-heating the gas increases its enthalpy (adding to Q_{in}) but also raises its temperature and specific volume and will therefore demand more work to compress a greater volume of gas. Compression is neither isentropic nor isothermal and must be polytropic where, assuming it acts as an ideal gas, PV^n = Constant, or $P = CV^{-n}$.

$$W = \int PdV = \int_{V1}^{V2} CV^{-n}\, dV = C\left[\frac{V_2^{-n+1}}{-n+1} - \frac{V_1^{-n+1}}{-n+1}\right] = C\left[\frac{V_2^{-n}V_2 - V_1^{-n}V_1}{-n+1}\right]$$

$$but\ C = PV^n\ and\ V^n \times V^{-n} = 1, therefore, W = \frac{P_2V_2 - P_1V_1}{-n+1}\ or$$

$$\boldsymbol{W = \frac{P_1V_1 - P_2V_2}{n-1}}$$ **Equation 3-12**

Two problems arise if the equation is to be used - what is the polytropic index 'n' and what is the volume change in the compression process? One might add that work input is also wasted as heat loss from the compressor during the compression process. The polytropic index 'n' is greater than 1 but less than gamma (γ) where gamma is the ratio of specific heats of the refrigerant at constant pressure and constant volume (Cp/Cv). The ratio of specific heats for R12 at 1.013 bar and 30°C is 1.13889[4], so, although specific heat varies with temperature and pressure, we will assume the polytropic index lies between 1.01 (Lenz, 2002)[5] and 1.14 since the isothermal index equals 1.

Another equation for polytropic compression (Cengal & Boles 1994 p345), which relies on the polytropic index but does not require the volume change is:

$$\boldsymbol{W_{comp} = \frac{nRT_1}{n-1}\left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$ **Equation 3-13**

For R12, the gas constant (R) = 68.7625 J/kgK[6]

[4] http://encyclopedia.airliquide.com/Encyclopedia.asp?GasID=22 [accessed 10 January 2014]
[5] Lenz provides a polytropic index for R12 of 1.091104
[6] www2.cc.kyushu-u.ac.jp/scp/system/library/PROPATH/manuals/p-propath/r12.pdf [accessed 10 January 2014]

Let $R = 0.069$ kJ/kgK, $P_1 = 300$ kPa, $P_2 = 2400$ kPa, $T_1 = 273$ K and the index ‘n’ lie between 1 and 1.14. The resulting compression work is given in Table 3-1.

Condition	n	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.1	1.11	1.12	1.13	1.14
0°C	W	-39.4	-39.8	-40.2	-40.6	-41.0	-41.4	-41.8	-42.2	-42.6	-43.0	-43.3	-43.7	-44.1	-44.5
5°C	W	-40.2	-40.6	-41.0	-41.4	-41.8	-42.2	-42.6	-43.0	-43.4	-43.8	-44.1	-44.5	-44.9	-45.3
10°C	W	-40.9	-41.3	-41.7	-42.1	-42.5	-42.9	-43.3	-43.7	-44.1	-44.5	-44.9	-45.3	-45.7	-46.1
ΔW @ 5K	W	-0.7	-0.7	-0.7	-0.7	-0.8	-0.8	-0.8	-0.8	-0.8	-0.8	-0.8	-0.8	-0.8	-0.8
ΔW @ 10K	W	-1.4	-1.5	-1.5	-1.5	-1.5	-1.5	-1.5	-1.5	-1.6	-1.6	-1.6	-1.6	-1.6	-1.6

Table 3—1 Compression work as a function of the polytropic index ‘n’

At 0°C compression work ranges between 39.4 and 44.5 kJ/kg as a function of the polytropic index, a difference of 5 kJ/kg; work done on the system is assigned, as would be expected, a negative value. At 5°C the additional work is between 0.7 and 0.8 kJ/kg whilst at 10°C the compression work increases by 1.4 to 1.6, Table 3-1. Changes in superheat of 5K and 10K result in an additional 1.8% to 3.7% increase in compressor work across the potential range of polytropic indices, Table 3-2.

Condition	n	1.01	1.02	1.03	1.04	1.05	1.06	1.07	1.08	1.09	1.1	1.11	1.12	1.13	1.14
Δ% @ 5K	W	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%	1.83%
Δ% @ 10K	W	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%	3.66%

Table 3—2 Impact of compression work as a function of the polytropic index and superheat

Whilst such increases appear to be marginal, the additive impact of each inefficiency raises the total entropy generated during the vapour compression cycle lowering the potential coefficient of performance. The control of superheat is a balance between minimizing superheat to reduce compression work, against ensuring full evaporation to protect the compressor from liquid “shoot through” leading to “liquid hammer” and damage to the compressor. Controlling the compression process is critical to achieving the highest coefficient of performance.

Control of refrigerant flow is achieved through the expansion valve (separating the high pressure side of the cycle from the low) where its primary function is to control refrigerant flow into the compressor in the form of gas. Fluid entering the compressor must be fully evaporated to ensure that there is no liquid to damage the compressor due to its incompressibility (liquid ‘shoot-through’) and to ensure this, the refrigerant is superheated before entering the compressor inlet. The simplest throttle device is a capillary tube as used in a refrigerator. Liquid entering the capillary tube flashes to gas and a liquid-gas mixture enters the evaporator where it is evaporated to produce a set value of superheat. Refrigerators may be modelled as steady state appliances since the internal source and sink temperatures (the fridge and kitchen temperatures) remain reasonably steady throughout the year. However, heat pump dynamic load will require variable controlled expansion, which can only be achieved with an adjustable expansion valve capable of responding to changes in source temperature whilst maintaining the correct level of superheat. Thermostatic expansion valves (TXV or TEV) are commonly fitted to domestic heat pump systems where the superheat temperature is used to control the flow of liquid into the evaporator through a closed loop control mechanism, Figure 3-13 (Danfoss). Bulb pressure Pb acts on a spring loaded diaphragm in the valve. As superheat increases or decreases changes in Pb provide feedback to adjust the flow rate of refrigerant into the evaporator, thus maintaining a constant level of superheat.

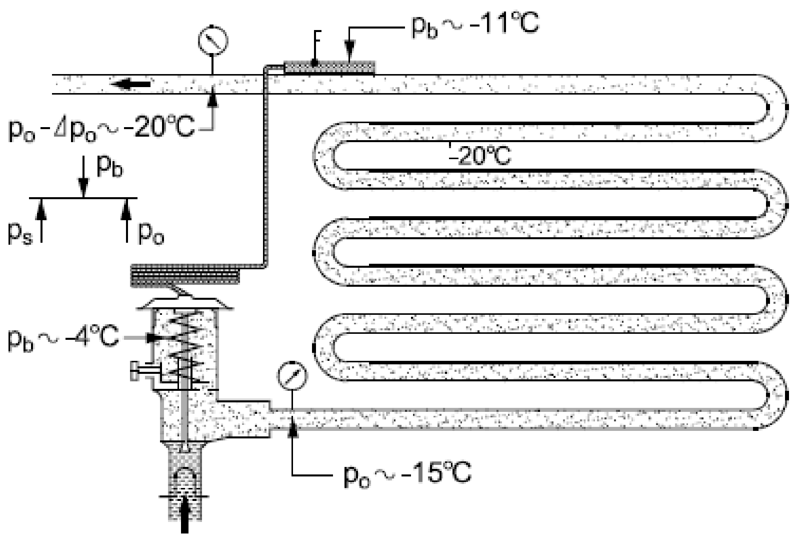


Figure 3—13 Thermostatic expansion valve (Source Danfoss)

High evaporator heat flux will cause rapid vapourisation and increased superheat. Whilst this provides heat into the cycle, superheating the vapour, as we have seen, also raises its specific volume and its temperature. Control of superheat is best described by recourse to manufacturers' literature. Al Maier (Maier, undated) states:

> "When superheat is too high, the liquid refrigerant is fully evaporated long before it reaches the evaporator outlet. As a result, the temperature of the refrigerant vapor continues to rise, raising the superheat of the gas in the suction line from the evaporator to the compressor. For every one degree rise in the suction gas temperature entering the compressor, there is a corresponding one and a half degree rise in the discharge gas temperature. This can lead to poor system performance and overheating of the compressor."

Compressor work, as defined by Abtar Singh (Singh, undated), is an inverse function of suction gas density:

$$\boldsymbol{W_c = \dot{m} \cdot P_s \cdot \frac{n}{n-1} \cdot \frac{1}{\rho_s} \cdot \left[\left(\frac{P_d}{P_s} \right)^{n-\frac{1}{n}} - 1 \right]}$$ **Equation 3-14**

where, W_c = Compressor power, $\dot{m}$ = mass flow rate (kg/s), P_s = pressure at suction (Pascals), P_d = pressure at discharge (Pascals), n = polytropic index and ρ_s is suction density (kg/m^3).

Singh comments: "Based on equation [Eq^n 3-14], lower suction return gas temperature results in higher density gas and hence lower compressor power. Too high a return gas superheat is therefore as bad as too low a return gas superheat."

Thermostatic expansion valves are adjusted to a set superheat for a pre-determined ambient operating point, typically (-5) to (-7)°C for an air source heat pump. However, due to its non-linear response (Figure 3-14), it is unable to keep a set superheat across the range of source temperatures (Bruderer, et al, 2008).

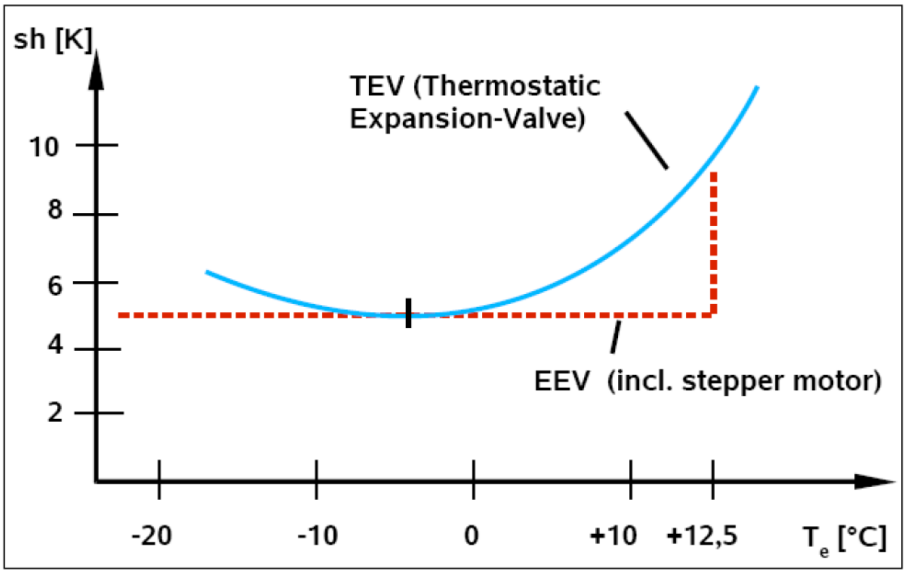


Figure 3—14 Superheat as a function of outside temperature for air-water heat pump. (Bruderer, 2008)

Bruderer suggests the potential for almost 15% increase in seasonal efficiency by replacing thermostatic with electronic expansion valves (EEV) due to their ability to respond to changes in conditions. Manufacturer's literature (Danfoss, Honeywell, Hanson, EEC, Alco, etc), propose superheat control through digital control signals fed by temperature/pressure transducers to, typically, pulse-width modulating electronic expansion valves and that deviation from set point can be corrected by integral and derivative control functions. Only three Japanese heat pumps in the UK EST trials, two Daikin and one Mitsubishi, use EEVs.

Sub-cooling condenser fluid ensures total heat transfer to the heating system and liquid exiting the condenser. This prevents flash gas from entering the thermostatic expansion valve, causing pressure oscillation, 'valve hunting' and thus the loss of superheat control. Control of condenser heat transfer is generally by on/off control of the compressor although a minority of heat pump manufacturers now use variable speed compressors to provide variable compression to better match dynamic loads, commonly with inverter control. The same two Japanese manufacturers featured in the trials also use inverter driven compressors supported with electronic expansion valves.

Domestic Hot Water

Carnot efficiency shows that the higher the condensing temperature the less efficient the heat pump, but domestic hot water (DHW) needs to be stored at 60°C plus if legionella is to be avoided. Figure 3-15 shows the impact of reducing condensing temperature for R407C, a common field trial refrigerant, from 70 to 60 to 50°C.

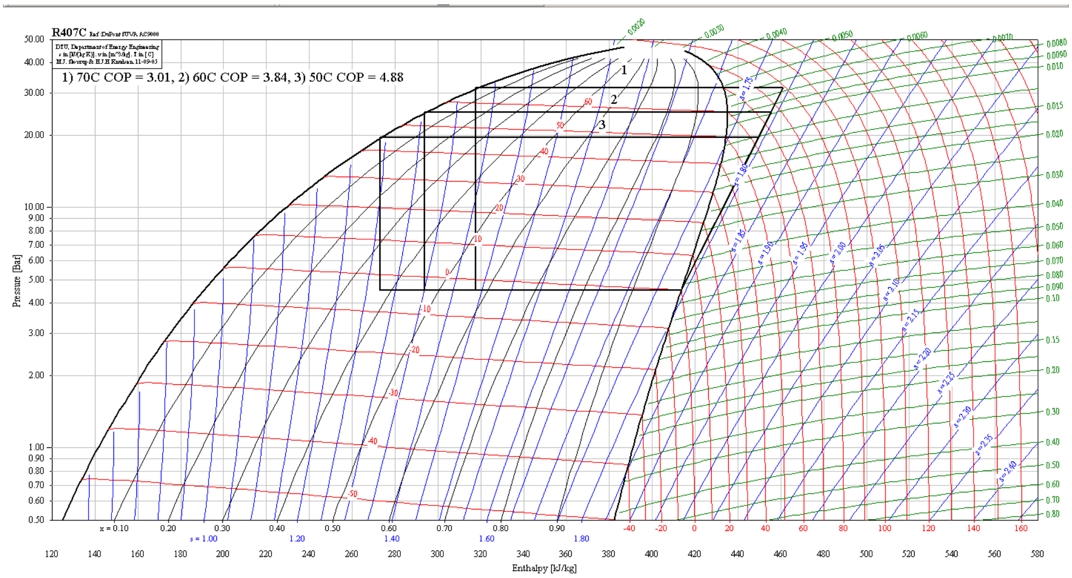


Figure 3—15 COP as a function of condensing temperature (R407C)

For isentropic compression, the COP of the heat pump increases at (1) from 3.01 to (2) at 3.84 and (3) at 4.88 as the condensing temperature drops from 70 to 60 to 50°C. A similar increase in efficiency associated with decreasing condensing temperatures can be observed in Figure 3-16 for R404A, again another trial heat pump refrigerant, where COP rises from 1.98 to 4.27.

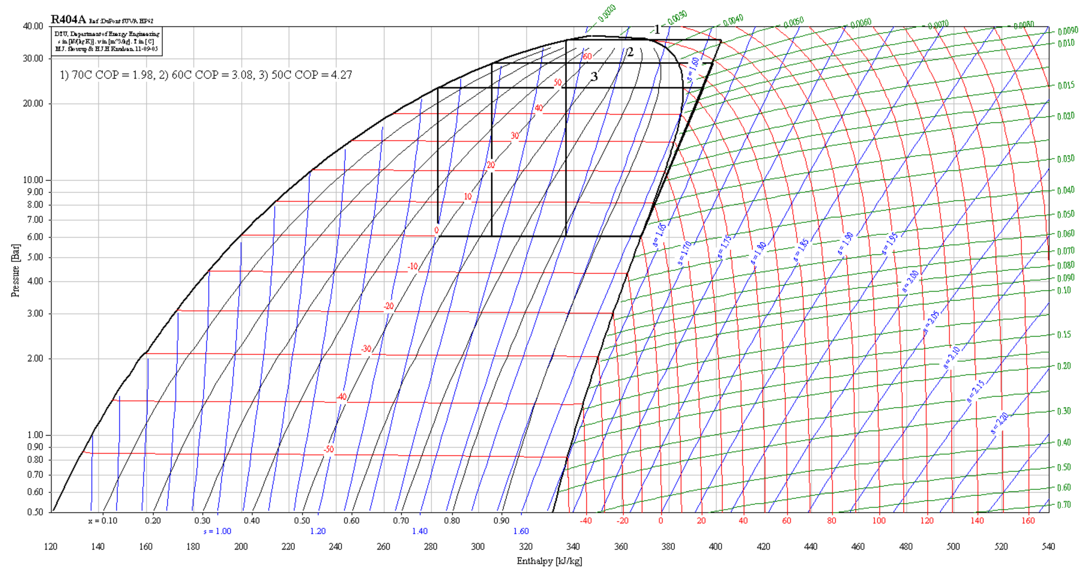


Figure 3—16 COP as a function of condensing temperature (R404A)

Heat pump manufacturers of HFC-based units set a limit of between 50 to 60°C for condensing temperature to maintain reasonable working COPs. For heat exchange to occur, there must be a temperature difference between the condenser and hot water storage, generally resulting in heat pump DHW storage temperatures of 50 to 55°C.

Desuperheating

A solution to low DHW temperatures, without resorting to resistance heating, may be provided by the process of 'desuperheating'. The Ph diagram, Figure 3-17, highlights the desuperheating process (off-compressor superheat to isothermal condensation) in two typical cycles for R410a, yet another trial refrigerant. Whilst most manufacturers limit condensation temperature to around 55°C, we see that the superheat temperature at this condition ranges between 90 and 55°C. A domestic hot water heat exchanger at the compressor outlet is able to exchange heat at these higher temperatures, resulting in hotter water without the use of resistance heating. Even with a low condensing temperature of 40°C associated with low temperature radiators or even underfloor heating, superheat ranges from 70 to 40°C, allowing a domestic hot water boost at a higher COP than resistance heating. Only one manufacturer of a desuperheating heat pump is featured in the EST field trials, the Global Energy 'London Eco Air Boiler', although there is no separate monitoring to the desuperheater function.

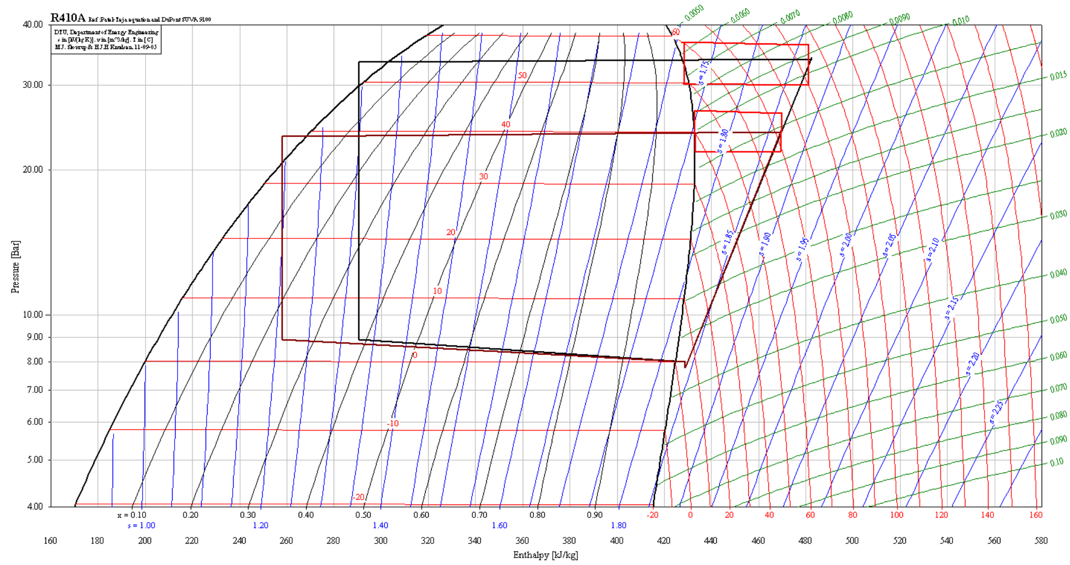


Figure 3—17 Desuperheater (shown using R410A) with 40 and 55°C condensing

Transcritical cycles

Hydrofluorocarbon refrigerants (HFC) have high global warming potential (GWP), with typical domestic heat pump refrigerants such as R134a, R404A, R407C and R410a having GWPs of approximately 1400, 3900, 1600 and 1700 (DuPont online) respectively, whereas carbon dioxide has a GWP of 1. Carbon dioxide (R744), however, has a critical point temperature of only 31°C and therefore requires a vapour compression cycle which operates above the critical point in order to have a practical application in heat pumps – such a cycle is described as ‘transcritical’; Figure 3-18 shows two such cycles. In the transcritical process heat out occurs in a ‘gas cooler’ rather than condenser. What is evident from the cycle is that superheat temperatures of 100°C are perfectly possible. For domestic hot water only, cycle 1’ – 4’ ensures that gas cooler outlet temperature operates at a minimum of 60°C. No transcritical systems feature in the field trials.

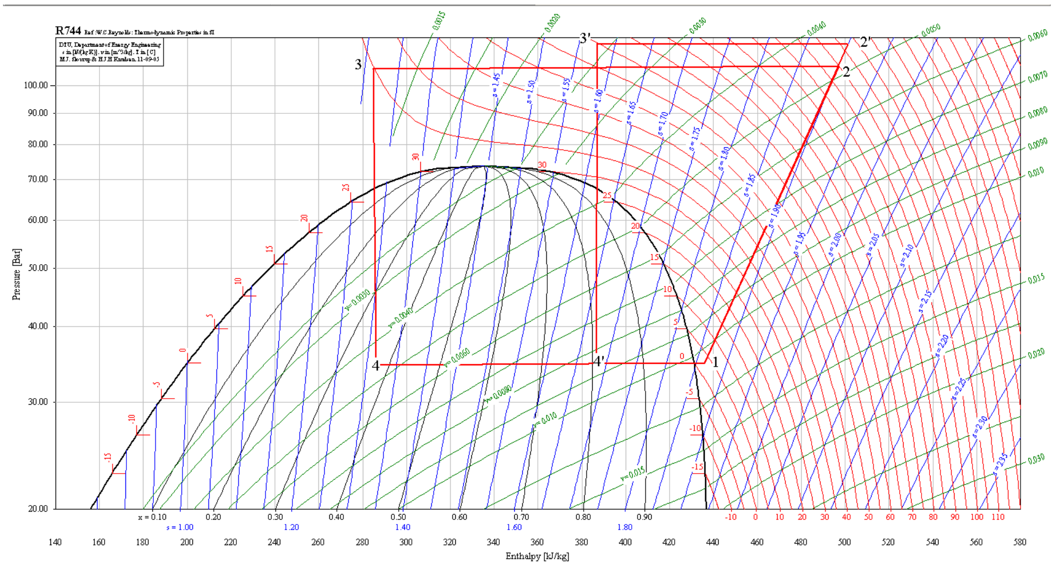


Figure 3—18 Transcritical cycles for CO2

System efficiency – a macro view

Real heating systems are a complex matrix of heat flows where the steady state model of heat pump COP no longer applies; the matrix of heat resistances and capacitances found in both the building and its heating system more closely matching a sixth or greater order partial differential equation (Levermore, 1997). Blomberg (1996) provides a full mathematical description of these three-dimensional heat flows and their differential equations. A purely visual assessment of these heat flows is shown in Figure 3-19.

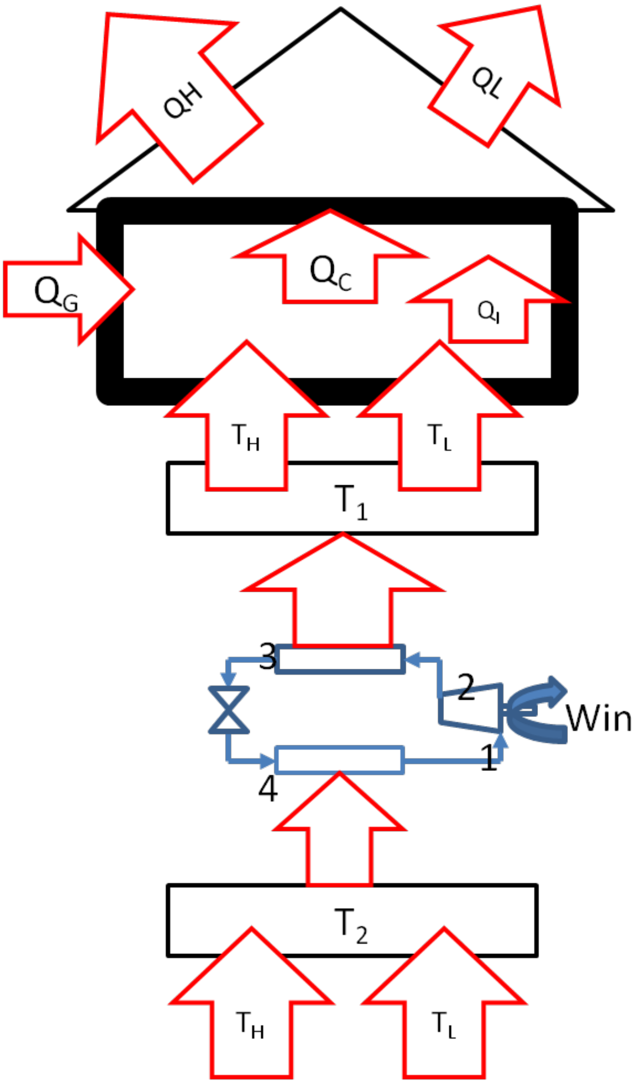


Figure 3—19 Modelling the dynamic factors that impact on COP

The cold reservoir T_2 may be subject to a large or small annual temperature variation depending on ground, water or air temperatures and diurnal variation in the case of air source heat pumps, T_H and T_L. Each of these variations will impact on the rate of evaporation since heat flow is a function of temperature difference. To respond to the variation in T_2, the evaporator pressure must be either fixed to the lowest possible evaporation temperature (clearly a major inefficiency) or be able to respond by varying both expansion and compression. For air source heat pumps, raising the minimum evaporation temperature will result in the need for less de-frosting, in cold damp weather, leading to higher heat transfer efficiency but potential failure during periods of low ambient temperature when heat is really needed.

Consider the hot reservoir T_1. The temperature requirements for T_1 depend on the system load which is itself under multiple influences such as heat flow into the structure, Q_C, during pre-heat (dependent the thermal mass effect or admittance) and where heat pump output must increase if reasonable pre-heat times are to be achieved due to this capacitance. Since the final emitter size will be fixed (the "radiators"), the only alternative to long pre-heat times, is higher compression, leading to higher condensing temperatures and higher mean water temperatures – all of which will lower the cycle COP.

Heat loss from the building, thermal transmittance, is driven by fabric and infiltration heat losses [$\Sigma UA + (\rho C_p NV)/(s/h)$], or W/K, and the variation in inside to outside temperature difference (ΔT). Whilst infiltration rate may vary depending on ventilation regime, the main driver is daily and seasonal temperature difference across the envelope and thus the heat output will need to vary from Q_L to Q_H. In addition to building admittance and transmittance, demand from the heat pump will be affected by the variation in internal and solar heat gains, Q_i and Q_G respectively. Where control of the vapour compression cycle is on/off, to respond to perhaps the occasional need for high temperature flow, the heat pump must be set to maximum compression and condensing temperature leading to minimum COP and high system parasitic heat losses from compressor, valves and pipes. Fixed speed compressor motors will begin to cycle or "hunt" as demand diminishes leading to short motor runs with high starter currents increasing work into the cycle. Alternatively the cycle may be fitted with a variable speed compressor leading to a higher overall COP as condensing temperatures rise and fall to meet demand.

It is evident that the coefficient of performance of a heat pump will vary during its operation depending on the relative impact of these multiple influences, the sophistication of the cycle components and its consequent ability to respond effectively. The heating system control regime may be on/off or weather compensated, where on/off control generally has a constant flow temperature and weather compensation allows for reduced flow temperature as heat losses diminish. Poorly insulated buildings with high heat losses will require high temperature emitters simply because low temperature emitters such as underfloor heating have insufficient output to offset heat loss. In addition to envelope losses, a standard central heating system will need to heat domestic hot water supplied at heat pump maximum output temperature. The lower COPs associated with high sink temperature are still likely to be greater than that of the most common alternative, the immersion heater, which has a maximum COP of only 1. Thus we find that the coefficient of performance for any single heat pump is dependent not only on the design of the vapour compression cycle, but:

- the variation in source temperature,
- the thermal characteristics of the building where it is installed
- the local climatic variation in outdoor temperature and indoor comfort set point

- emitter design temperature
- the ratio of space to water heat demand
- and hot water storage temperature

Since heat pump coefficient of performance varies over time, the most applicable equation for COP measured in the field must be Equation 3-8, but expressed as a ratio of heat and work "rates", Equation 3-15.

$$\boldsymbol{COP_{HP}} = \frac{\dot{\boldsymbol{Q}}}{\dot{\boldsymbol{W}}} = \frac{\boldsymbol{J/s}}{\boldsymbol{J/s}} \qquad \text{Equation 3-15}$$

where $\dot{Q}$ is the rate of heat flow from the heat pump and $\dot{W}$ is the rate electrical work input. Equation 3-15 can be integrated over a short or long time period to assess instantaneous, monthly or seasonal performance. Heat pump annual efficiency is known as its Seasonal Performance Factor (SPF), Equation 3-16.

$$\boldsymbol{SPF} = \frac{\boldsymbol{kWh/yr_{out}}}{\boldsymbol{kWh/yr_{in}}} \qquad \text{Equation 3-16}$$

Summary

The chapter has introduced the concepts of heat, work and heat engines. The early work of Carnot, Thompson and Clausius provided definitions for reversibility, thermodynamic temperature and the generation of entropy as a measure of irreversibility and gave rise to a definition for the second law of thermodynamics – that is, maximum potential cycle efficiency is a function of the ratio of reservoir absolute temperatures T_L and T_H. Where the objective is cooling, the cycle is described as a refrigerator whereas when it is heating the cycle is a heat pump. Importantly, Rogers and Mayhew state that for real (irreversible) cycles, T_L and T_H represent the respective temperatures of the fluid media transferring heat into and out of evaporator and condenser and thus the potential efficiency is directly related to choice of heating system flow temperature. Whilst entropy provides a theoretical assessment of irreversibility, its use as a practical measure of system efficiency is limited, with little current literature describing its use in calculating efficiency "in the field" (apart from the term isentropic efficiency used to describe compressor efficiency, Equation 3-11). Refrigerant manufacturers provide pressure-enthalpy rather than temperature-entropy diagrams for all commercial refrigerants.

Work-in, provided by compression, is required to produce heat output, but the heat output demand for a domestic heat pump will vary during operation. In order to control fluid flow in the cycle, protect the compressor from liquid 'shoot through' and minimise compressor work, variable control over expansion is achieved with thermostatic or electronic expansion valves to ensure full vapourisation but also to minimise superheat. Heat pumps used for space heating and hot water have dynamic source temperatures and dynamic heat loads. Domestic heat pump compressors are generally single speed, on/off controlled but inverter driven models are available where motor speed is variable so that compression may adjust for dynamic loads leading to a closer match to system demand, less motor starts and lower parasitic loads associated with "hunting" and lower starting amps. Dynamic loads are best suited to electronic expansion valves with their ability to respond to changes in evaporator condition through temperature/pressure transducer fed pulse-width modulation. Close control may be achieved with proportional, integral and derivative (PID) controllers that respond rapidly and remove any offset from superheat set point, functions of derivative and integral control respectively. Inverter driven compressors and advanced controls provide energy savings but at additional cost and added complexity. It is seen that the dynamic nature of the vapour compression cycle as it responds to changes in load means that cycle efficiency is dependent on its particular application rather than solely on the manufactured design – any particular heat pump will have a different efficiency depending on a range of site specific variables. There is also the implication of the theoretical requirements for reversible operation that systems tend to become more reversible the slower they operate. Reductions in the speed of a system can be compensated for without reducing useful heat output, by increasing the time the system operates. In the limit this favours systems that operate continuously to satisfy demand.

Whilst heat pump COP may be described by a laboratory test at steady state conditions with fixed temperature differentials across evaporator and condenser, real heat pump COP is subject to continuous change and demands a metric such as "seasonal coefficient of performance" (SCOP) or seasonal performance factor (SPF) to recognise this fact. SCOP or SPF is a function of the interactions between the heat pump and the system as it responds to dynamic operation. Any additional inputs to the system, such as electrical resistance heating or circulation pumps, create yet other boundaries and the need to further differentiate. The next chapter will investigate how these differences are measured.

Chapter 4 Assessing COP and SPF

Introduction

The previous chapter provides a thermodynamic definition of coefficient of performance (COP), whether reversible or irreversible, that acts as a theoretical concept. A further, or regulatory, definition of heat pump COP is provided by product manufacturers based on EN 14511 laboratory testing at fixed conditions. As such, a manufacturer's COP test does not represent real world installations, which function at different source and sink temperatures during their annual operation and where the heat pump could be attached to both space heating and domestic hot water systems with different temperature requirements and therefore different values of COP. The use of COP by manufacturers in advertising literature leads to a general misunderstanding by the public of heat pump capabilities. Several methods for testing both space heating and DHW COP are described including that carried out by the author at the Barrett Green House, BRE, UK. The process of monitoring is itself fraught with potential hazards that require the monitoring designer to understand the mechanics of the heat pump, the monitoring equipment and the data calculations required to assess performance. Some understanding of the role of weather on air source heat pump seasonal efficiency is provided by the "bin method" approach proposed in EN 15316, where COP data is aligned with local weather data to provide a seasonal coefficient of performance (SCOP). Whilst this method provides a mathematically derived annual performance, it remains an approximation since the COP associated with each bin is derived from EN 14511. This chapter is thus a search for a "real" seasonal efficiency or, in the case of heat pumps, "seasonal performance factor" (SPF). SPF provides a metric for comparing the "as-installed" efficiency of one heat pump central heating system to another. An ostensibly simple demand for comparison of different systems at the design stage introduces a number of challenges in applying EN 14511 test results and in defining an appropriate 'system boundary' for analysis and comparison between different forms of heating. The research has attempted to model performance using EN 14511 test data and the mathematical analysis based on the EN 15316 "bin method", giving rise to what is best described as the seasonal coefficient of performance (SCOP).

This chapter therefore has four main aims: to describe the requirements of laboratory testing coefficient of performance for space heating; to describe the additional complexity of assessing COP for domestic hot water based on the literature and pilot study results; to describe the bin method for deriving a seasonal coefficient of performance (SCOP) and to outline the limitations of applying such methods.

Note: where the source of a translation into English is not acknowledged, the reader may safely blame it on the author.

Coefficient of Performance

The test regime for coefficient of performance is described in EN 14511:2007 comprising of four parts under the general title "Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling“:

- Part 1: Terms and definitions
- Part 2: Test conditions
- Part 3: Test methods
- Part 4: Requirements

The standard provides a laboratory test regime based on fixed source and sink temperatures for water and air source heat pumps. The standard provides the single descriptor of “water-to-water” for both ‘ground-to-water’ and ‘water-to-water’ heat pumps since both use water as the heat transfer fluid. In practice the water is treated with antifreeze (commonly ethylene or propylene glycol) and is known as “brine”. Air source heat pumps have either outdoor air or indoor exhaust ventilation air as the source. Testing may be under “Standard rating conditions” or “Application rating conditions” as shown in Tables 4-1 and 4-2. Whilst standard rating conditions express the COP for a particular heat pump at a single source and sink temperature, the minimum information required under the standard, application rating conditions provide COP values across a range of source and sink temperatures typical of those encountered in heat pump system design.

		Outdoor heat exchanger		Indoor heat exchanger	
		Inlet temperature °C	Outlet temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Water	10	7 [a]	40	45
	Brine	0	-3 [a]	40	45
	Water (for floor heating or similar application)	10	7 [a]	30	35
	Brine (for floor heating or similar application)	0	-3 [a]	30	35
Application rating conditions	Water	15	[b]	[b]	45
	Brine	5	[b]	[b]	45
	Brine (for floor heating or similar application)	5	[b]	[b]	35
	Brine	- 5	[b]	[b]	45
	Brine	0	[b]	[b]	55
	Water	10	[b]	[b]	55

[a] For units designed for heating and cooling mode, the flow rate obtained during the test at standard rating conditions in cooling mode (see Table 8) is used.
[b] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

Table 4—1 Water-to-water test conditions (EN 14511-2: 2007)

		Outdoor heat exchanger		Indoor heat exchanger	
		Inlet dry bulb temperature °C	Inlet wet bulb temperature °C	Inlet temperature °C	Outlet temperature °C
Standard rating conditions	Outdoor air	7	6	40	45
	Exhaust air	20	12	40	45
	Outdoor air (for floor heating or similar application)	7	6	30	35
Application rating conditions	Outdoor air (for floor heating or similar application)	2	1	[a]	35
	Outdoor air (for floor heating or similar application)	- 7	- 8	[a]	35
	Outdoor air (for floor heating or similar application)	- 15	-	[a]	35
	Outdoor air	2	1	[a]	45
	Outdoor air	- 7	- 8	[a]	45
	Outdoor air	- 15	-	[a]	45
	Outdoor air	7	6	[a]	55
	Outdoor air	-7	-8	[a]	55

[a] The test is performed at the flow rate obtained during the test at the corresponding standard rating conditions.

Table 4—2 Air to Water test conditions. (EN 14511-2: 2007)

Testing air source heat pumps at low source temperatures can lead to temperatures in the air surrounding the evaporator falling below dew point. For the refrigerant to extract heat, the temperature difference between the source and refrigerant may lead to evaporator surface temperatures below freezing and thus the build-up of ice. Although the initial wetting will enhance heat exchange, ice build up insulates the evaporator heat exchanger and requires a de-frosting cycle and therefore a loss of output. Icing is a particular problem for air source heat pumps at low temperature and/or high humidity and requires COP testing to incorporate a transient cycle that includes defrost. The European Heat Pump Association provide extended guidance (EHPA, 2009[a,b & c]) on heat pump testing and with their publication on air to water testing, in particular, provide a detailed description of transient testing.

In order to standardise manufacturers' information, the EHPA demand ASHPs are tested to all of the test temperatures in Table 4-3.

Test conditions / testing points air/water heat pumps:							
			Source			Sink	
Test cond.	Standard	Type[1]	inlet temperatures			outlet temperatures	
			T.db (°C)	T.wb(°C)	φ (%)	T.out (°C)	T.in (°C)
A7/W35	EN 14511-2	N	7	6	89	35	30
A2/W35[2]	**EN 14511-2**	**QL**	**2**	**1**	**84**	**35**	**a**
A-7/W35	EN 14511-2	B	-7	-8	75	35	a
A-15/W35	EN 14511-2	B	-15	--	--	35	a
A7/W45	EHPA	B	7	6	89	45	a
A7/W55	EHPA	B	7	6	89	55	a
A-7/W55	EHPA	B	-7	-8	75	55	a
A20/W55	EHPA	B	20	14	50	55	a

1 Type name: N -> standard rated point, B -> operating rated point, QL -> point to determine COP for Quality Label
2 Relevant test condition for the EHPA Quality label

a The test is conducted with the volume flows indicated at A7/W35

Table 4—3 Conditions for performance of Air to Water COP testing. (EPHA, 2009a)

Steady state conditions for source and sink must not deviate from those shown in Table 4-4. Measurements outside of these limits lead to prolonged testing under transient conditions described below.

4.5.1 Test conditions during heating operation – steady state

Measured quantity:	Permissible dev. (±) of the arithmetic mean values from set value	Permissible dev. (±) of individual measured values from set values
Air:		
Temperature, dry-bulb	0,3 K	1 K
Temperatue, wet-bulb	0,3 K	1 K
Relative humidity[1]	3 % RH	7 % RH
Water/brine:		
Inlet temperature	0,2 K	0,5 K
Outlet temperature	0,3 K	0,6 K
Volume flow	2%	5%
Electrical voltage	4%	4%

Table 4—4 Permissible deviations from set values during steady testing (EPHA, 2009a)

Steady state testing is described by the EHPA as follows:

> "Start the measurement with a preconditioning period, during which the actual values over at least ten minutes must lie within the tolerance limits defined in [Table 4-4] [Region A Figures 4-1 and 4-2].
> Follow this by a defrost cycle with a 10-minute recovery phase with the defrosting being triggered automatically or manually by the test item control gear. During this period,

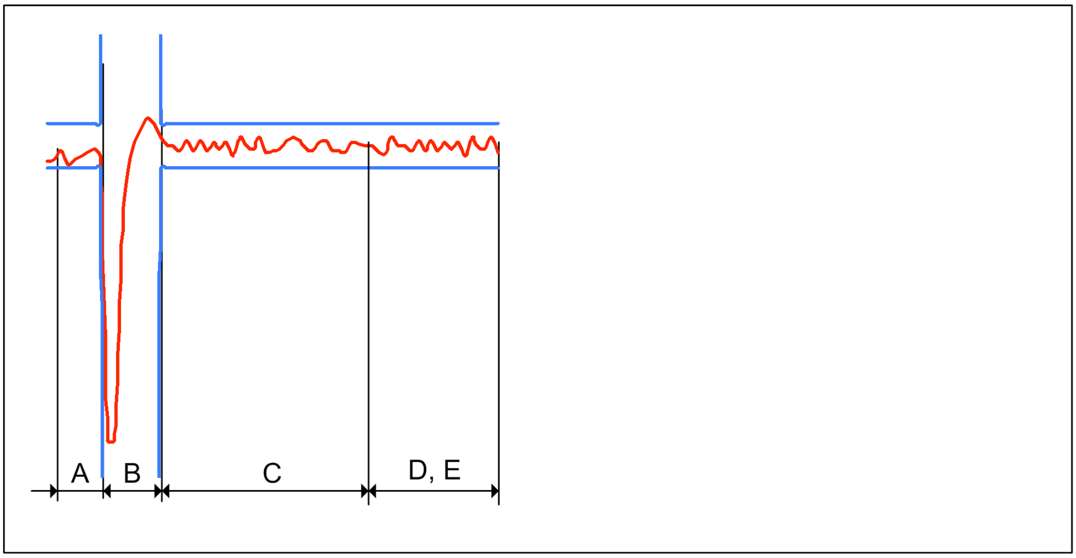
deviations from the desired values as shown in [Table 4-4] are permissible [Region B, Figures 4-1 and 4-2].

Once the recovery phase has ended, the actual values must again be within the tolerance limits given in [Table 4.4]. The equilibrium phase that follows lasts 60 minutes [Region C Figures 4-1 and 4-2]. Follow this by the data collection period, which last[s] 35 minutes or three hours, depending on the test conditions.

If defrosting occurs during the equilibrium phase and/or the measurement period, the tolerance limits for the desired values during the defrost phases and the subsequent 10-minute recovery period according to [Table 4-5] apply, defrost cycle test conditions – transient.

Record the measured values every 10 seconds throughout the entire measurement period."

It can be seen from Figure 4-1 that the steady state data collection period is 35 minutes.



A Preconditioning period: 10 minutes
B Defrost with recovery time: defrost time + 10 minutes
C Equilibrium period: 60 minutes
D Data collection period: 35 minutes
E Period for capacity calculation

Figure 4—1 Steady state testing without defrosting (EPHA, 2009a)

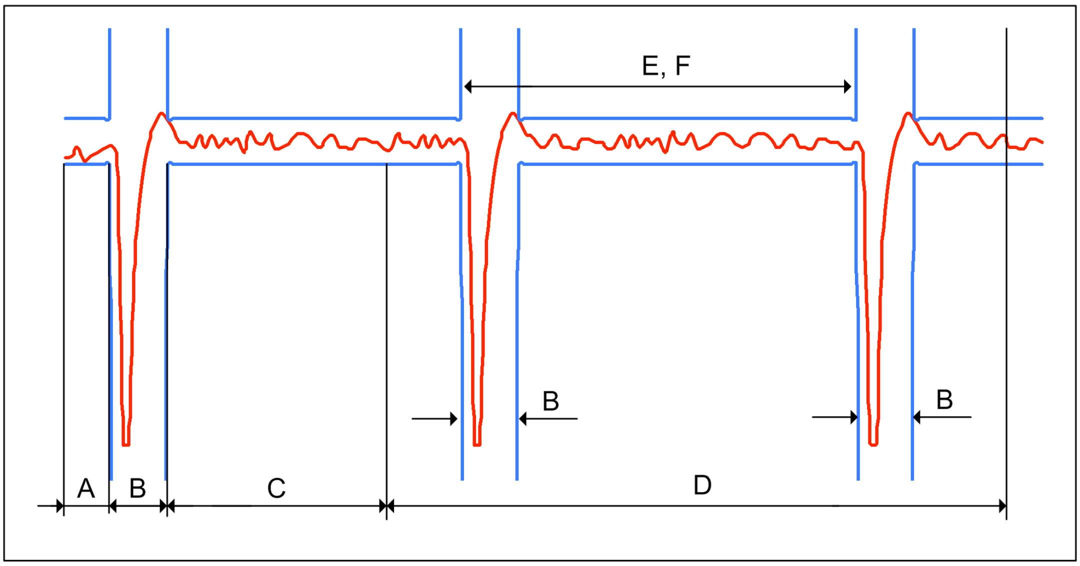
"Transient test conditions are present if one or more defrosting cycles occur during the equilibrium or data collection period. The defrosting processes of the evaporator are triggered only by the defrost control of the test object".

Transient conditions are shown in Table 4-5 and: "apply during a defrost cycle and during the first 10 min after the termination of a defrost cycle when the heat pump is operating in the heating mode." The EHPA document suggests that transient conditions apply to all tests, "with an air temperature less or equal +2 °C", (p11).

Measurement quantity:	Permissible dev. (±) of the arith- metic mean values from set value	Permissible dev. (±) of individual measured values from set values value
Air:		
Temperature, air entering indoor-side:		
dry-bulb	1,5 K	2,5 K
wet-bulb	-	-
Temperature, air entering outdoor-side:		-
dry-bulb	1,5 K	5,0 K
wet-bulb	1,0 K	-
Relative humidity[1]	9 % RH	-
Water/brine:		
Inlet temperature	-	-
Outlet temperature	-	± 2 K
Volume flow	2%	5%
Electrical voltage	4%	4%

Table 4—5 Transient permissible deviations from set values during testing (EPHA, 2009[a])

The transient process is shown in Figure 4-2 where it is apparent that the data collection period "D" is 3 hours and may cover multiple defrosting cycles.



A Preconditioning period: 10 minutes
B Defrost with recovery time: defrost time + 10 minutes
C Equilibrium period: 60 minutes
D Data collection period: 3 hours
E Period for capacity calculation
F Work cycle with defrost time

Figure 4—2 Transient testing with multiple defrosting cycles during data collection (EPHA, 2009[a])

As can be seen in Table 4-2, air to water heat pump testing to EN 14511:2 requires manufacturers to supply just a single "standard rating conditions" test at 7/6°C (dry and wet bulb) source to 35/30°C sink. Ground-to-water and water-to-water heat pumps minimally require a single test at 0/(-3)°C (source flow into the heat pump and return to the ground) to a 35/30°C sink such as underfloor heating, Table 4-1. EN 14511 does not describe any form of transient testing for water-to-water heat pumps. The EHPA's guide to Testing of Water/Water and Brine/Water Heat Pumps (EPHA, 2009[b]) makes no reference to defrosting or transient operation and perhaps leads to the assumption that, although the source return temperature is at (-3)°C and therefore the evaporator refrigerant temperature at least some 2 or 3 degrees lower, there will be no defrost. EN 14511 Table 1 lists water-to-water heat pumps as "units designed for installation indoors" at temperatures between 15 and 30°C. The test implies that ground and water source heat pumps will be housed in a heated space with sufficiently low relative humidity to prevent dew point wetting and subsequent freezing. This is certainly not the case with the EST field trials where a number of externally housed ground source heat pumps are identified. Equally, high internal wet bulb temperatures, a common enough phenomenon in the UK, will result in condensation on cold surfaces such as the evaporator.

Testing facilities

The schematic drawing (BRE, 2007[a]) in Figure 4-3 describes the Building Research Establishment's heat pump test chamber, designed for test certification to EN 14511. Source air temperature and humidity are stabilised using proportional, integral and derivative (PID) controls whilst buffer vessels with heat rejection, again PID controlled, provide the sink.

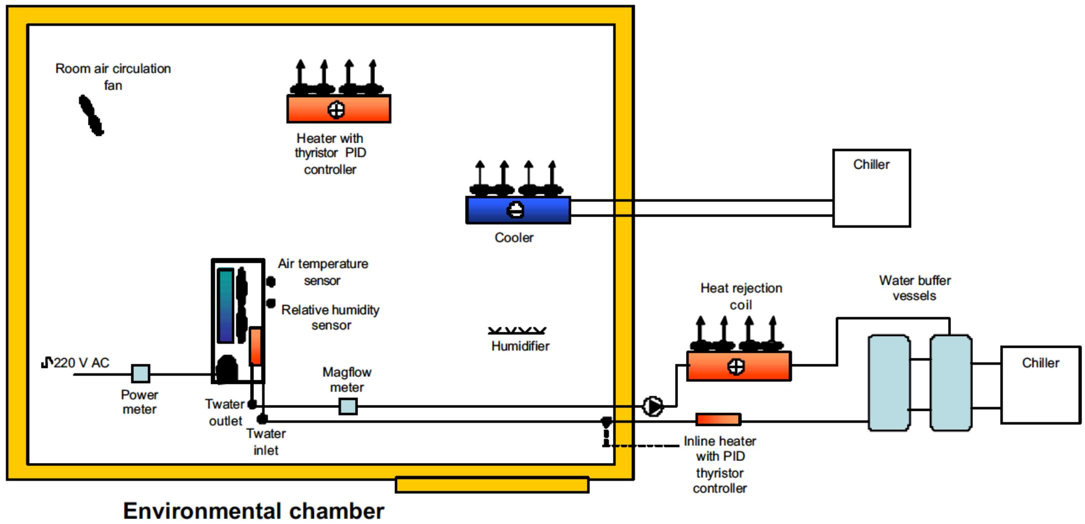


Figure 4—3 BRE EN 14511-3 Test Chamber (BRE, 2007[a])

A similar set up is shown in Figure 4-4, the Building Services Research Information Association's test chamber (BSRIA, 2009). The BSRIA schematic also identifies differential pressure transducers on the heating flow and return.

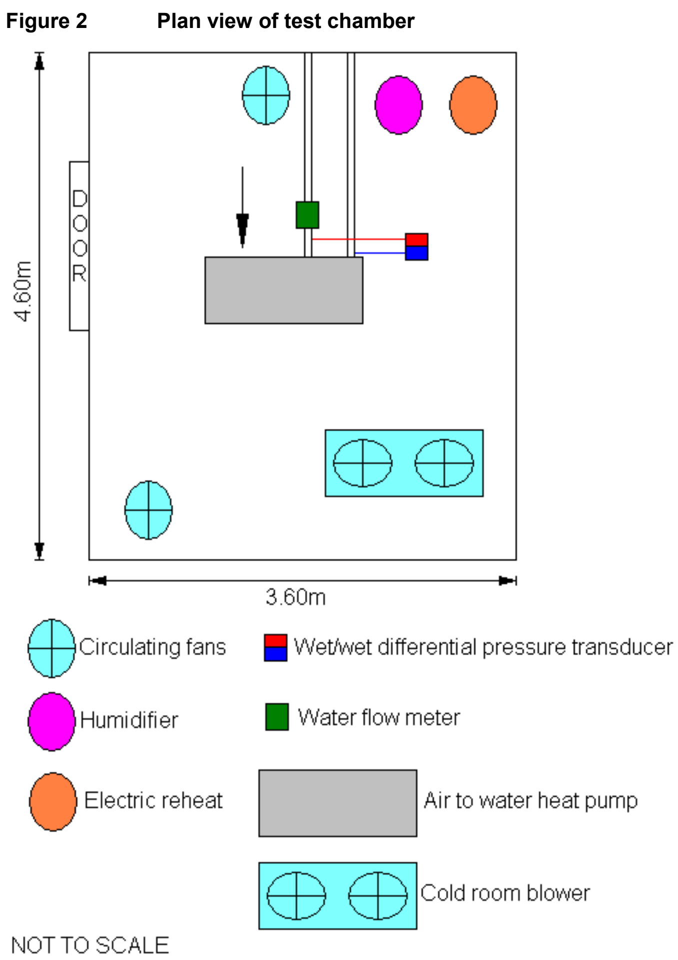


Figure 4—4 Plan view of BSRIA ASHP test chamber. Ref BSRIA, 2009

Fans and pumps

For a heat pump to operate, there must be a flow of source fluid over the evaporator. For ASHPs this is in the form of a fan and therefore electrical power input must be the sum of fan, compressor and controls. Water-to-water heat pumps require the circulation of source water over the evaporator, whilst both air to water and water-to-water heat pumps require a circulator (a central heating pump) to drive heat exchange from the condenser and circulate it through the heating system. However, the energy demand of pumps will depend on mass flow rate of fluid and the index circuit resistance of the pipework and is therefore variable between one installation and another. EN 14511 resolves this by measuring the pressure drop associated with the heat exchangers and adding an allowance for the power requirements of these losses only, based on a rule of thumb approach of assuming the pump is 30% efficient, Figure 4-5.

4.1.6.1 If a liquid pump is an integral part of the unit, only a fraction of the input to the pump motor shall be included in the effective power absorbed by the unit. The fraction which is to be excluded from the total power absorbed by the unit shall be calculated using the following formula:

$$\frac{q \times \Delta p_e}{\eta} \text{ [W]} \qquad (6)$$

where:

— η is 0,3 by convention;

— Δp_e is the measured available external static pressure difference, in Pascals;

— q is the nominal water flow rate, in cubic meters per second.

4.1.6.2 If no liquid pump is provided with the unit, the proportional power input which is to be included in the effective power absorbed by the unit, shall be calculated using the following formula:

$$\frac{q \times \Delta p_i}{\eta} \text{ [W]} \qquad (7)$$

Figure 4—5 Power input of liquid pumps (EN 14511-3)

The EPHA endorse this methodology:

> "The effective power consumption of the heat pump can be calculated from the power consumption of the blower, the compressor and all electrical mechanisms of the heat pump that function during heat mode. The power consumption figures for the delivery apparatus of the heat pump are considered only to such an extent that they are required to overcome the internal static pressure differentials." (EHPA 2009[a]).

The assumption of 30% efficiency may be an overestimation. Nørgard, et al (1983, p18) in their Danish study on pumps and pumping, state of typical standard circulation pumps:

> *"For små cirkulationspumper i centralvarmeanlæg m.v., der i dag bruger cirka 2% af landets el-forbrug, er den samlede effektivitet af pumpe og motor sjældent over 10%. Der er således teoretisk basis for en forbedring af pumpernes effektiviteter."*
>
> "For small circulation pumps in central heating systems, etc., which today amounts to about 2% of the country's electricity consumption, the overall efficiency of pump and motor rarely rises above 10%."

Since 1983 there have been two EU Directives which impact on pumping: Energy-using products 2005 (European Commission, 2005) and Ecodesign requirements for energy-related products 2009 (European Commission, 2009). Under these Directives, new energy efficiency standards have been set which introduce energy labelling (A-G) and a measurement known as the EEI or energy efficiency index (a ratio of the actual energy use to a reference energy use – hence the lower the EEI the better).

Manufacturers have responded with new pump designs which can work either as a more energy efficient fixed speed pump or, what appears to be the most common high efficiency pump, the permanent magnet variable speed design.

Since the Directives have driven the market change there is a great deal of literature covering new pump design, including the report to the Commissioners, Lot 11 - 'Circulators in buildings' (Faulkner, 2008). Lot 11 outlines the methodology and test results for low energy pumps. Remarkably, it states that a typical conventional domestic circulator has an overall efficiency of 18% (p50).

A report for the British Pump Manufacturers' Association (BPMA) suggests:

> " A high efficiency circulator in variable speed mode consumes only 76 kWh per year instead of 287 kWh per year, which is a saving of 73 %", (p4). (Bidstrup & Seymour, undated).

The Phase 2 EST Heat Pump report (Dunbabbin, et al, 2013) describes the use of high efficiency pumps "low energy dc pumps" which indicates permanent magnet with inverter drive, what they also call: "variable speed DC pumps". The results show:

> "systems with variable speed DC pumps had, on average, a lower difference between SPFH3 and SPFH4 (0.06 as opposed to 0.13)," (p35).

This would appear to be about a 50% saving on pumping – the only difference between SPF_{H3} and SPF_{H4}.

However, irrespective of the pump operating efficiency, the system volume flow rate (mass flow rate to carry the heating demand) and hydraulic design remain important to overall pumping efficiency. A system must be designed to match the design heat loss, therefore carry the optimum amount of water at the right temperature. In addition, a high system pressure loss will always demand a higher pumping power than one designed for lower pressure loss. Thus the impetus is still on the designer understanding the role of mass flow rate, pipe sizing/pipe diameter, pressure loss per metre pipe run and valve and fitting velocity pressure loss factors (zita factors).

BSRIA EN 14511 test report

Some indication of the relative importance of circulation pumps in manufacturer testing can be found from an EN 14511 air source test provided by UK BSRIA test laboratory for the Eco Tec EAS 10kW unit (BSRIA, 2009):

> "The following are the results from the Eco Tec Heat Pumps Ltd EAS1000M air to water heat pump test complete with CoP and duty. The pump power value was calculated in accordance with clause 4.1.6.2 of EN14511-3:2007, because no liquid pump was provided with the unit."

A summary of the test outputs is provided in Figure 4-6. Note that the average water flow rate and pressure difference across the condenser during the test are 0.44 l/s and 6.15 kPa respectively. Applying EN 14511 clause 4.1.6.2, (0.00044 x 6150/0.3 = 9.02) the power associated with overcoming the pressure drop through the condenser is just a 9 Watt addition to the 2,656 Watts for fan, compressor and controls; its omission leading to a potential error of just one third of one percent. Even at 10% efficiency, the EN 14511 pump allowance is just 27 Watts or 1%. For field trial comparison, one needs to consider that many manufacturers supply integrated pumps for both ground and air source heat pumps. Where ground loop pumps are metered through a single heat pump meter, there is the obvious difficulty of extrapolating the evaporator pressure drop only, the same applies to both ground and air source sink circulators. One could also argue that for internally mounted pumps motor energy is converted to heat and is thus a useful output. Such nuances complicate the assignment of power demand and heat output when designing monitoring systems.

HEATING TEST SUMMARY A7/6, W30/35

			Units
1	Atmospheric Pressure	101.57	kPa
2	Ambient Temperature	20.99	°C
3	Electrical Quantities		
	Voltage to unit	232.03	V
	Total current to unit	13.8	A
	Total power to unit	2.656	kW
	Pump power	0.009	kW
	Effective power input to unit	2.665	kW
4	Thermodynamic Quantities		
a	Outdoor heat exchanger		
	Air inlet temperature, dry bulb	6.98	°C
	Air inlet temperature, wet bulb	6.05	°C
	Air inlet relative humidity	87.60	%
b	Indoor Heat Exchanger		
	Water inlet temperature	30.00	°C
	Water outlet temperature	34.82	°C
	Water flow rate	0.440	l/s
	Pressure difference	6.15	kPa
5	**Heating Capacity**	8.862	kW
6	**CoP**	3.33	-
7	Test Period		
	Date of test	22-Sep-09	
	Duration of test (steady state)	35.0	minutes

Figure 4—6 Average outputs from EN 14511 test at 7/35 (BSRIA,2009)

BRE EN 14511 test report

The UK Building Research Establishment provide a comprehensive EN 14511 test report for the Mitsubishi Ecodan inverter driven air to water heat pump (BRE, 2007[a]). The report provides COP data for 100% and 50% outputs at compressor speeds 7 and 4 respectively in line with the EN 14511 approach to variable speed compressors. Test data is provided for supply air at (-5), 2/1, 7/6 and 12/10°C dry/wet bulb and sink temperatures of 35/30, 45/40 and 55/50°C. These outside air temperatures of (-5) to 12°C provide a reasonable range to model annual UK weather conditions and heating systems operating in continuous (24 hour) mode where there is no preheat condition. The maximum air temperature of 12°C is reasonably close to a low energy building balance point, traditionally 15.5°C, where internal gains match heat losses. Sink temperatures are reflective of underfloor heating and a range of low and medium temperature radiators.

Test	Inlet/outlet water temperature	Air dry bulb / wet bulb temperature	Compressor speed step	Date of test	Power input	Heat output	COP
1	30/35°C	7/6°C	7	16/2/07	2350.0	8800.5	3.74
2	30/35°C	7/6°C	4	19/2/07	1370.4	5842.7	4.26
3	30/35°C	2/1°C*	4	1/3/07	1790.5	5775.7	3.23
4	30/35°C	2/1°C*	7	22/2/07	2861.2	8005.3	2.80
5	30/35°C	-5°C*	4	14/3/07	2112.5	5797.3	2.74
6	30/35°C	-5°C*	7	15/3/07	3178.3	7784.1	2.45
9	40/45°C	7/6°C	4	20/2/07	1756.6	6134.7	3.49
10	40/45°C	7/6°C	7	20/2/07	3056.1	9290.5	3.04
11	40/45°C	2/1°C*	4	27/2/07	2281.7	5989.4	2.62
12	40/45°C	2/1°C*	7	28/2/07	3554.2	8296.3	2.33
13	40/45°C	-5°C*	4	8/5/07	2521.8	5578.6	2.21
14	40/45°C	-5°C*	7	15/3/07	3916.1	8048.0	2.06
17	50/55°C	7/6°C	4	21/2/07	2339.6	5911.2	2.53
18	50/55°C	7/6°C	7	21/2/07	3832.8	8901.3	2.32
19	50/55°C	2/1°C*	4	5/3/07	2832.4	5978.8	2.11
20	50/55°C	2/1°C*	7	8/3/07	4245.6	8245.6	1.96
21	50/55°C	-5°C*	4	9/5/07	3181.5	5284.2	1.66
22	50/55°C	-5°C*	7	27/3/07	4840.3	7281.5	1.50
25	30/35°C	12/10°C	4	29/3/07	1209.9	5923.3	4.90
26	30/35°C	12/10°C	7	30/3/07	2083.2	9050.9	4.34
27	40/45°C	12/10°C	4	2/4/07	1802.4	6440.9	3.57
28	40/45°C	12/10°C	7	31/3/07	2719.3	9015.6	3.32
29	50/55°C	12/10°C	4	3/4/07	2343.7	6329.1	2.7
30	50/55°C	12/10°C	7	3/4/07	3943.0	9824.2	2.49

*At the 2/1°C and -5°C conditions the evaporator coil frosted-up and a transient test method was employed (see BS EN 14511-3 Clause 4.5.3.2.)

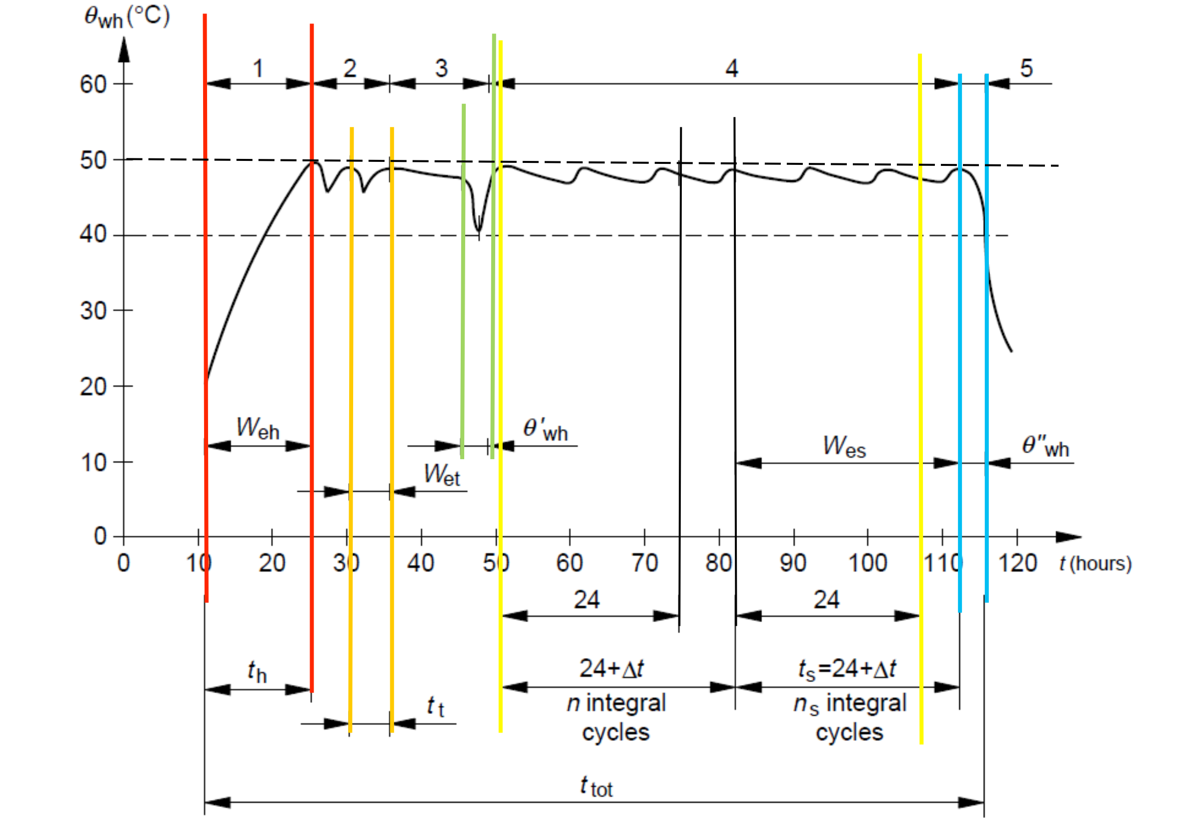
Figure 4—7 Mitsubishi Ecodan heat pump rating test results (BRE 2007[a])

The test results indicate that a 50% reduction in compressor speed results in increased efficiency at all supply temperatures ranging from 12 to 15% at 30/35°C and 8 to 11% at 50/55°C, Figure 4-7. The ability to provide modulating output should also, in theory, result in less heat pump cycling.

Domestic hot water production

As well as space heating, domestic heat pumps are required to produce domestic hot water (DHW). The test regime for DHW is described by EN 255-3:1997, and more recently by EN 16147:2011. These standards provide COPs which are applicable to a specific heat pump incorporating a specific cylinder of set size and heat loss, supplying data on standby losses and various tapping regimes.

EN 255-3 calculates only the "Tapping COP" (COPt). The tapping COP is shown in Figure 4-8 between the orange coloured lines, whilst the section between the red lines represents the heat up period.



1 Heating up period

2 Determination of the coefficient of performance for tapping sanitary hot water

3 First part of the determination of the reference water temperature

4 Determination of standby power input

5 Second part of the determination of the reference water temperature and determination of the maximum quantity of usable hot water

NOTE. The average temperatures, θ'_{wh} and θ''_{wh}, are specified in **4.6.4**.

Figure 4—8 DHW test heating regime (EN 255-3)

The heating up period (*th*) is described thus:

> "The test is started with the entire contents of the storage tank at the temperature of the ambient air specified in table 3 [15°C]. This is achieved by circulating the water until the temperature at the outlet is within the limits for the ambient.... It shall be ensured that the entire heat pump system is in thermal equilibrium with its surroundings. The heat pump is switched on. The heating up time, *t*h, is measured from the time the heat pump is switched on until it is shut off by the hot water thermostat situated in the tank. The heating up energy input, *Weh*, is determined over the same period as the heating up time."

Coefficient of performance (COP tapping) is measured as follows:

1. Cylinder thermostat turns off after heat up period. Start tapping at 0.2 dm^3/s, compressor starts and runs. Tap a total of 0.5 V_n (nominal volume). Measure till cylinder thermostat switches compressor off.
2. Repeat step 1. Cylinder thermostat switches off for third time. Energy content of the two draw offs not to differ by >10%.
3. Measure tapping and reheating time (t_t) between compressor off 2 and off 3.
4. Tapping flow rate (q_{wh}) plus incoming cold water (Θ_{wc}) and outgoing hot (Θ_{wh}) measured at least every 10 seconds. The tapped hot water energy is calculated for temperatures >40°C as:

   $$Q_t = \int_0^{tt} \rho wh \,.\, Cpw \,.\, qwh \,.\, (\Theta wh - \Theta we)\ dt$$

   Where ρwh density of HW at flowmeter, Cpw average water specific heat at constant pressure, qwh flow rate of HW, $(\Theta wh - \Theta we)$ temp diff between HW outlet and CW inlet secondary water, dt integration time in seconds of the procedure.
5. Reheating energy input (W_{et}) determined over final tapping period.
6. COP_t is calculated as $COPt = \frac{Q_t}{W_{et} - P_{es}\ .t_t}$

   Where W_{et} reheat energy, P_{es} is the standby energy losses/standby time and t_t is the duration of the standby test. COP_t may be seen as a measure of energy out over energy in, Equation 21:

$$\boldsymbol{COP_t} = \frac{\frac{kg}{m^3} \times \frac{J}{kgK} \times \frac{m^3}{s} \times K \times s}{J - \left(\frac{J}{s} \times s\right)} = \boldsymbol{J/J}$$ Equation 4-1

It is necessary to measure standby losses. Allow the cylinder to cycle, with no draw off, for not less than 24 hours and at least 3 cycles. Time the standby duration. Energy input for standby

$$P_{es} = \frac{W_{es}}{t_s}$$

Where W_{es} takes into account fans and pumps according to Sections 4.2.8 and 4.2.9

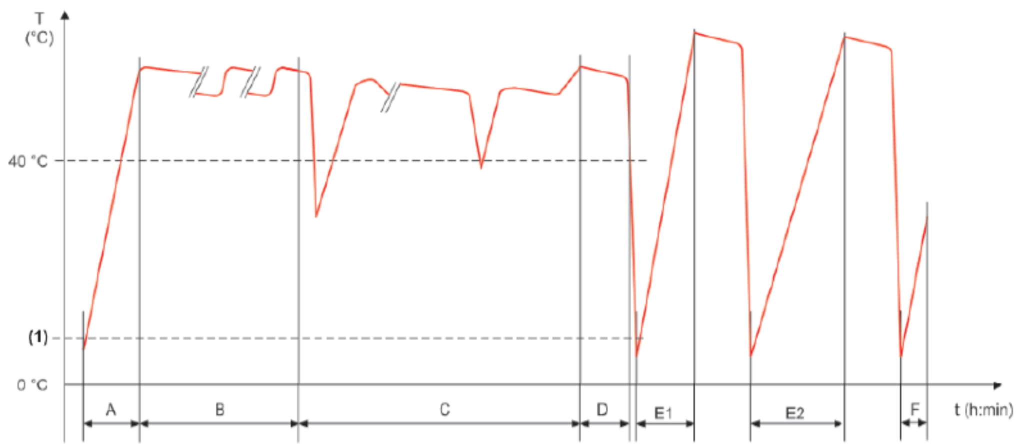
EN 255-3 provides the following information, Figure 4-9:

Reference hot water temperature (θ_{wr})	°C
Coefficient of performance for tapping hot water (COP_t)	
Maximum quantity of hot water in a single tapping (V_{max})	dm^3
Heating up time (t_h)	h and min
Heating up energy input (W_{eh})	kWh
Standby power input (P_{es})	W

Figure 4—9 Presentation of main results (EN 255-3)

We note that the only COP result required is the COP for tapping. EN 255-3 does not require the publication of the COP for heating from cold. Note also that cylinder heat losses are subtracted from the COP denominator; the standard provides COP to heat the tapped hot water only.

EN 16147:2011 provides a similar test profile but allows the manufacturer to choose a tapping regime from a range of options, the test profile is shown in Figure 4-10 and the presentation of results in Figure 4-11. Note that "COP_{DHW}" is based on the specific tapping regime only, there is no requirement to publish the COP for heating from cold which, theoretically, should be higher.



Key

(1) cold water temperature

Figure 4—10 EN 16147 DHW test regime

A. Heating up period
B. Determination of standby power
C. Determination of energy consumption and COP for DHW by using one of the tapping cycles provided

D. Determination of reference DHW temperature and maximum quantity of useable water in a single tapping

E. Determination of the temperature operating range

F. Safety tests

Result	Unit
Heating up time	h:min
Heating up energy input	kWh
Standby power input	W
Class of the measured tapping cycle and the determined electrical energy consumption $W_{EL\text{-}TC}$ for each measured cycle	No. kWh
COP_{DHW} and class of used tapping cycle	-
Reference hot water temperature	°C
Maximum quantity of usable hot water	l
Temperature operating range: Minimal and maximal heat source temperature, minimal start and maximal mean temperature domestic hot water	°C

Figure 4—11 Presentation of DHW results (EN 16417)

Although five tapping regimes are provided, it would appear that manufacturers are testing to the tapping cycle described as XL in Figure 4-12. Evidence for this is provided by test certificates, Figure 4-13, from the *Warmepumpen Testzentrum (WPZ), Interstaatliche Hochschule fur Technik Buchs* (NTB) in Switzerland, which identify cylinder size, test regime and tapping COP.

	Start (h:min) Tapping cycle time	Energy (kWh)	Type	ΔT desired (K), to be achieved during tapping	Min. ΔT (K), start of counting useful energy
1	07:00	0,105	Small		15
2	07:15	1,82	Shower		30
3	07:26	0,105	Small		15
4	07:45	4,42	Bath	30	0
5	08:05	0,105	Small		15
6	08:15	0,105	Small		15
7	08:30	0,105	Small		15
8	08:45	0,105	Small		15
9	09:00	0,105	Small		15
10	09:30	0,105	Small		15
11	10:00	0,105	Small		15
12	10:30	0,105	Floor cleaning	30	0
13	11:00	0,105	Small		15
14	11:30	0,105	Small		15
15	11:45	0,105	Small		15
16	12:45	0,735	Dish washing	45	0
17	14:30	0,105	Small		15
18	15:00	0,105	Small		15
19	15:30	0,105	Small		15
20	16:00	0,105	Small		15
21	16:30	0,105	Small		15
22	17:00	0,105	Small		15
23	18:00	0,105	Small		15
24	18:15	0,105	Household cleaning		30
25	18:30	0,105	Household cleaning		30
26	19:00	0,105	Small		15
27	20:30	0,735	Dish washing	45	0
28	20:46	4,42	Bath	30	0
29	21:15	0,105	Small		15
30	21:30	4,42	Bath	30	0
Total		19,07			

Equivalent hot water at 60 °C 0,325 m^3

Figure 4—12 Tapping regime XL (EN 16417)

Prüfresultate Warmwasser-Wärmepumpen für Luft-Wasser basierend auf der EN 16147
Test results of domestic hot water heat pumps based on EN 16147

Auftraggeber Customer	Gerät Type	Prüfnummer Test number	Bauart Type of construction	Produktart Product type	Kältemittel Refrigerant	Kältemittelmenge [kg] Capacity of refrigerant	Nennvolumen [dm^3] Nominal volume	Aufheizzeit [h:min] heating up time	Aufheizenergieaufnahme [kWh] heating up energy input	elektrische Verlustleistung [W] standby power input	Bezugswarmwassertemperatur [°C] reference hot water temperature	max. nutzbare Warmwassermenge [dm3] max. useful volume of heat water	Entnahmezyklus tapping cycle	COP	COP (according to EN 255-3)	Schallleistungspegel [dB(A)] Sound power level
Atlis AG Schlössliweg 2-6 CH - 4500 Solothurn	Heatmaster CH-301	015-12-03	a	S	R410A	0.63	**300**	05:39	4.8	49	53	369	XL	**2.4**	3.0	**66**
Domotec AG Lindengutstrasse 16 CH - 4663 Aarburg	HPWH 250 SOL	013-12-01	a	S	R134a	1.28	**250**	05:18	3.8	41	54	315	XL	**2.8**	3.2	**67**
Kibernetik AG Langäulistrasse 62 CH - 9470 Buchs	WPLW-KIB-BW-300L	014-12-02	a	S	R134a	0.95	**300**	07:17	4.5	39	54	380	XL	**2.9**	3.3	**55**

Figure 4—13 EN 16147 Test results for tapping cycle XL. (WPZ)

As well as providing the tapping COP, the WPZ certificate also provides a number of interesting outputs for the three heat pump units including heat up time (from 5 hours and 18 minutes to 7 hours and 17 minutes) and the storage temperature rise (from 20°C to 53 or 54°C). All three cylinders are over 300 litres in size. We also note, based on the three units in Figure 4-13, that EN 16147 COP is lower by about 15% than that derived by EN 255-3.

EN 255-3 and EN 16147 provide a method for assessing just the "tapping COP" based on a half volume tapping and a specific tapping regime respectively. As such they are of limited use for understanding the DHW efficiency of a heat pump operating at non-EU standard tapping patterns over 24 hours and with a non-integrated cylinder typical of UK traditional central heating design. In addition, few manufacturers provide either EN 255-3 or EN 16147 data and, as noted, many provide just a single COP value at EN 14511 "standard rating conditions" of A7/W35 or B0/W35.

Where EN 14511 COP data is used to model DHW production at a fixed heat pump outlet temperature such as 45 or 55°C, it is based on a constant heat sink demand which does not reflect the gradually reducing temperature difference between the primary coil and stored cylinder water and thus the gradually declining heat transfer. Yet another anomaly is provided by the fact that some heat pump manufacturers set a maximum heat pump output temperature of around 50°C due to the decreasing efficiencies associated with nearing the critical point for the refrigerant. It would appear that none of the test standards provide sufficient data to accurately assess the probable seasonal efficiency for all heat pump driven hot water storage systems thus requiring a test programme for each heat pump with, for example, a typical domestic cylinder. Since cylinders come in different sizes, with different levels of insulation and different heat transfer coils, there is no standard cylinder. In addition, hot water use is extremely variable between households, not least because smaller storage and hotter water can be mixed with cold to provide sufficient hot water at the required temperature.

With regard to storage temperature, the UK Energy Savings Trust's report on hot water consumption (EST/DEFRA, 2008) Measurement of Domestic Hot Water Consumption in Dwellings states:

> "The mean delivery [draw off] temperature for regular boilers is 52.9° C with 95% confidence interval **±** 1.5° C."

The report also suggests that household daily hot water use is given by the equations in Figure 4-14 below.

	Whole dataset		5 occupants or fewer	
	Value	p-value	Value	p-value
Intercept (litres/day)	46 ± 22	4.4%	40 ± 24	9.7%
Slope (litres/person.day)	26 ± 7	0.0%	28 ± 7	0.0%
Consumption model	46 + 26 N		40 + 28 N	

Figure 4—14 Daily hot water consumption (EST/DEFRA, 2008)

Figure 4-15, showing the daily run off profile of the whole sample, indicates two main tapping periods of morning and evening. Unless the cylinder is cold, reheat will be from an intermediate temperature between cold feed and final storage temperature and therefore it is difficult to predict heat pump actual sink flow and return temperatures.

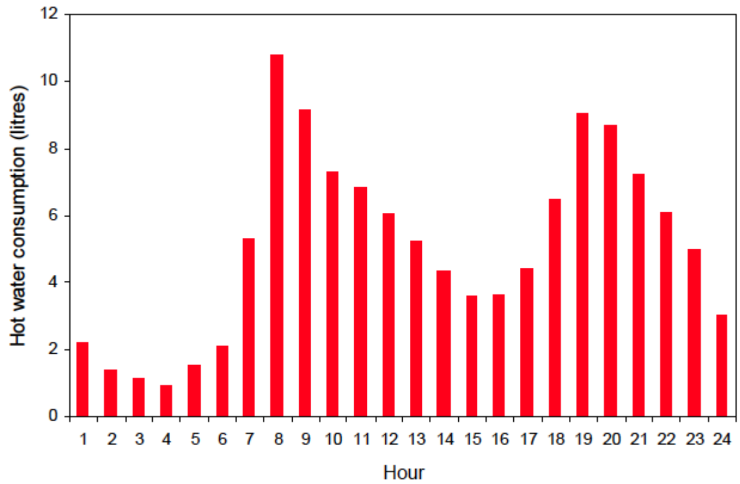


Figure 4—15 Daily DHW tapping profile (EST, 2008)

The European test centre WPZ provide examples for EN 16147 and EN 255-3 test results for a range of domestic hot water heat pumps. At the time of writing, none of the heat pumps on sale in the UK appear to refer to either of these British Standards/European Norms; results are given for EN 14511-3 space heating tests only. In order to assess DHW COP various other test regimes have been suggested.

BRE DHW methodology

The BRE test report for the Mitsubishi Ecodan (BRE, 2007[a]) provides an assessment of domestic hot water COP based on the test set-up shown in Figure 4-16.

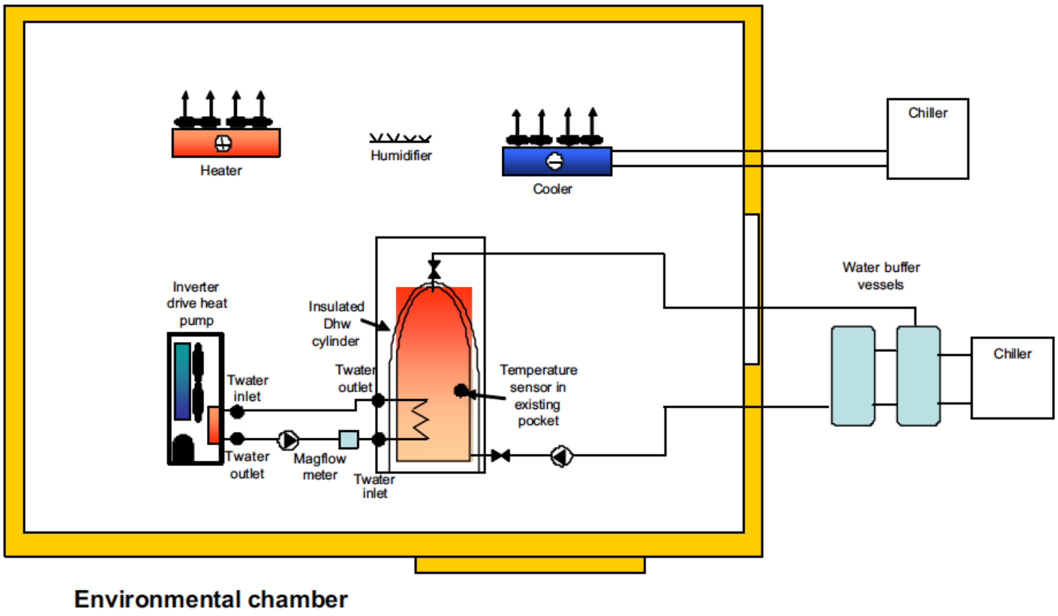


Figure 4—16 BRE DHW COP test (BRE, 2007)

The BRE report states:

> "Note that BS EN 14511 supersedes BS EN 255 which has been withdrawn [replaced in October 2011 by EN 16147]. A series of hot water cylinder tests were also carried [out]. Although there is no BS EN standard for these tests relevant parts of BS EN 14511: 2004 were used as a guide. The hot water cylinder was a propriety product supplied by Gledhill Water Storage Ltd."

The report describes heating capacity as equal to: $P_H = q \times \rho \times C_p \times \Delta T$,

where q = water volume flow rate

ρ = density of water

C_p = specific heat of water

ΔT= difference between water inlet and water outlet temperatures

and COP = ratio of heating capacity to effective power input of unit with EN 14511 allowances for pumping.

BRE state:

> "The DHW cylinder heat pump tests were based on heating the hot water cylinder from 12°C to 55°C. The electrical power input and the heat input to the cylinder were determined on the basis of their integrated values over the test duration."

What is described is a heat up test from cold; there is no tapping test. We see that the heating capacity (P_H) is based on the temperatures across the pumped secondary water draw off and cold feed. Note that there is no calculation of standby losses, perhaps because the test is specifically for the Glynwed cylinder. Tabulated results are provided, Figure 4-17.

Test	Air dry bulb / wet bulb temperature	Compressor speed step	Time to heat tank from 12°C to 55°C (secs)	Mean heat pump COP	Total power consumed by heat pump (kWh)	Total heat supplied to the tank (kWh)	Date of test
			Mean of 2 tests				
1	7/6°C	7	3338	2.87	2.62	7.53	27/7/07
2	7/6°C	4	5013	3.16	2.37	7.50	27/7/07
3*	2/1°C	7	3876	2.06	3.66	7.54	30/4/07
4	2/1°C	4	4529	2.70	2.80	7.57	30/4/07
5	-5°C	7	4188	1.79	4.24	7.57	1/5/07
6	-5°C	4	5502	2.00	3.81	7.62	2/5/07
9	12/10°C	7	3334	3.18	2.35	7.48	11/4/07
10	12/10°C	4	4932	3.63	2.07	7.50	10/4/07
11	20/16°C	7	3375	3.65	2.06	7.52	12/4/07
12	20/16°C	4	4530	4.25	1.77	7.51	3/5/07
13	25/18°C	7	3417	4.09	1.82	7.45	19/4/07
14	25/18°C	4	4736	4.90	1.51	7.39	20/4/07 & 3/5/07

*Test 3 - 1 test carried out

Figure 4—17 BRE DHW test results. Note description of Power in kWh (BRE, 2007[a])

Whilst maximum output at compressor speed 7 demands a greater compressor workload, and thus a lower COP, it does reduce heat up time by about 20 minutes. It is also worth stating that the secondary circulation (domestic hot water loop) is pumped with the circulation pump in the cold feed, Figure 4-16. This circulation of cylinder water will increase heat transfer from the primary coil in comparison to the heat transfer normally associated with temperature-stratified cylinders. Note that the cylinder thermostat, half way up the cylinder, confirms the assumption of forced circulation since the cylinder water temperature is given as 55°C throughout.

Importantly, based on the kWh data, the test results in Figure 4-17 confirm a lower energy demand when operating at Speed 4 (50%) rather than Speed 7 (100%) where the longer heat up time at Speed 4 results in a lower energy demand for the same DHW output and thus a higher COP.

BRE DHW graphical analysis

The full report, only available from Mitsubishi (BRE, 2007[b]), also provides graphical analysis of the DHW test runs in Appendix B. Examples are shown for Tank test 10 (run 2), at an ambient air temperature of 12°C, Figure 4-18, and Tank test 5 at (-5)°C, Figure 4-19.

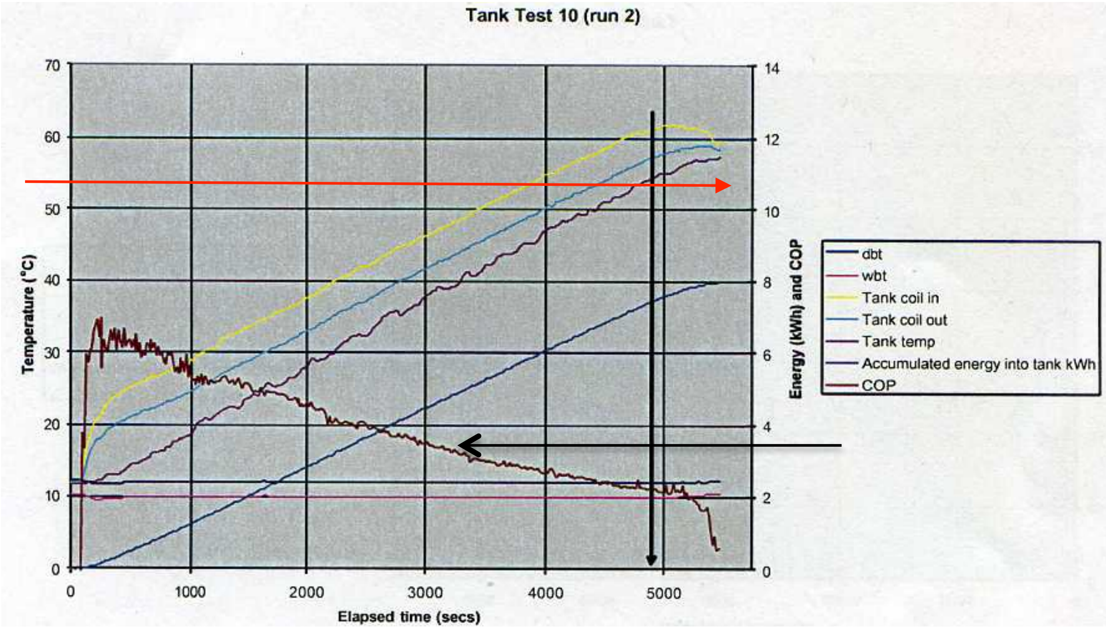


Figure 4—18 Tank Test 10: Ambient 12°C, cylinder temperature 55C, mean COP 3.63 (BRE, 2007[b])

Test 5 appears to show a defrost cycle with clearly visible reduced outputs at the primary flow and return, "tank coil in" and "tank coil out".

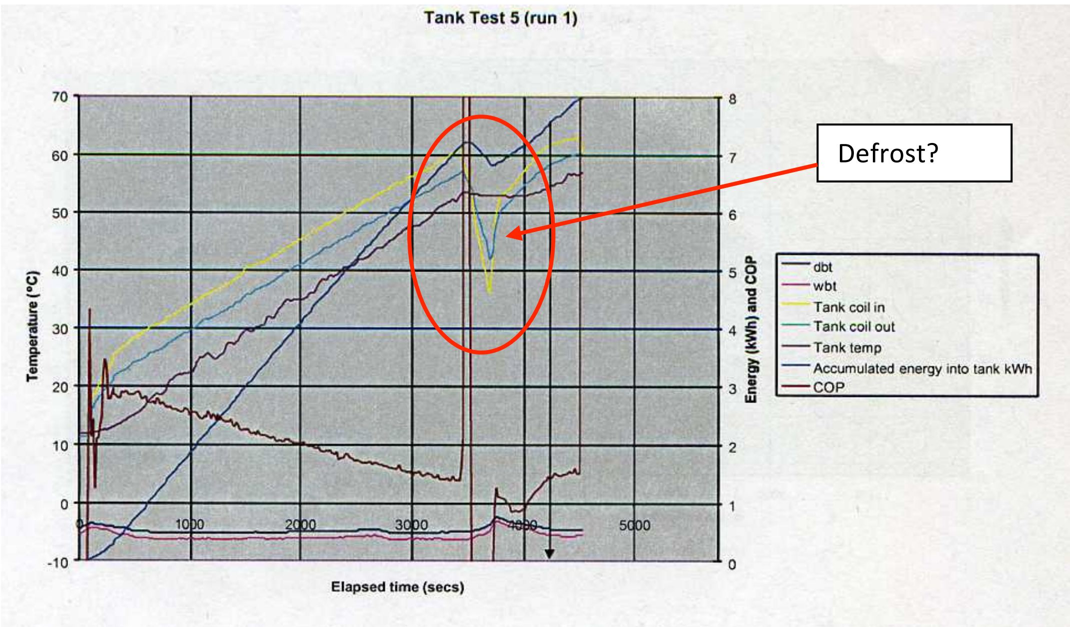


Figure 4—19 Tank Test 5: Ambient (-5)°C, cylinder temperature 55C, mean COP 1.79 (BRE, 2007[b])

Passivhaus DHW Methodology

The PH methodology for testing hot water production for Compact Units is given in:

Prüfverfahren zur energetischen und schalltechnischen Beurteilung von Wärmepumpen-

Kompaktgeräten für die Zertifizierung als "Passivhaus geeignete Komponente" (Passivhaus, 2007) or, in English, "Test methods for energetic and sound technical [acoustic] evaluation of heat pump compact units for certification as a 'Passive suitable component'". The test regime is based on EN 14511 for space heating with the proviso that the minimum test temperature is dependent on source air being supplied through underground ducting (thus pre-warming the air to a minimum of (-2)°C) and leading to testing at (-2), 2 and 7°C. Underground supply ductwork can be sized using the Passivhaus "PH*Luft"* software, available from the Passivhaus website. This is a test for a combined service appliance based on an MVHR unit with an air source heat pump, located in the extract duct after the heat exchanger and providing heat to the supply air and DHW coil. The combined unit has been the subject of an International Energy Agency heat pump study (Wemhoner & Afjai, 2006), Figure 4-20.

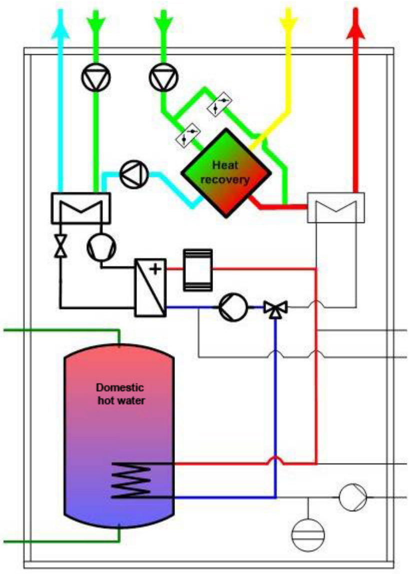


Figure 4—20 Compact unit with DHW testing (Wemhoner & Afjai, 2006)

Domestic hot water testing is based on the same supply temperatures of (-2), 2, 7, plus the additional 20°C for summer operation. The procedure includes heating from cold, *Speicheraufheizung*, "heating of store from cold" (from 20 to 50°C) and reheating following tapping, *Speichernachladung,* "recharging storage" (from 40 to 50°C). Subtractions are made for cylinder and primary pipework losses, EN 14511 additions for pumping requirements and, uniquely, DHW compressor heat loss providing additional space heating in winter. The test result is therefore applicable to a particular compact unit heat pump and cylinder combination. The following description is based on a test certification report for the Drexel & Weiss *Aerosmart M* compact unit prepared by the Hochschule Bremen: *Prüfbericht Prüfung eines Kompaktgerätes zur Zertifizierung als Passivhaus geeignete Komponente*, or, "Consideration of a test report for certification of a compact unit as a suitable Passivhaus component" (available from the manufacturer Drexel & Weiss).

The report states: *Die Umwalzpumpe sorgte wahrend der Prufungen fur eine permanente Durchmischung des Speichers, wodurch die Nutztemperatur genau bestimmt werden konnte.* This approximates as: "The circulation pump during the tests caused a permanent mixing of storage whereby the utility temperature could be determined accurately." It would appear that the COP heating from cold is based on using a shunt pump to circulate the cylinder water and is thus by forced convection. Whether this complies with standard German installation practice, the use of a DHW secondary return, cannot be established.

It also appears from the report that cylinder losses are subtracted from the Work-in as useful space heating and since the compressor also produces up to 200 Watts of heat, which can be utilized in the warm air heating during winter months, this additional heat output is added to the DHW output to enhance the overall COP for test temperatures of (-2)° C and 7°C.

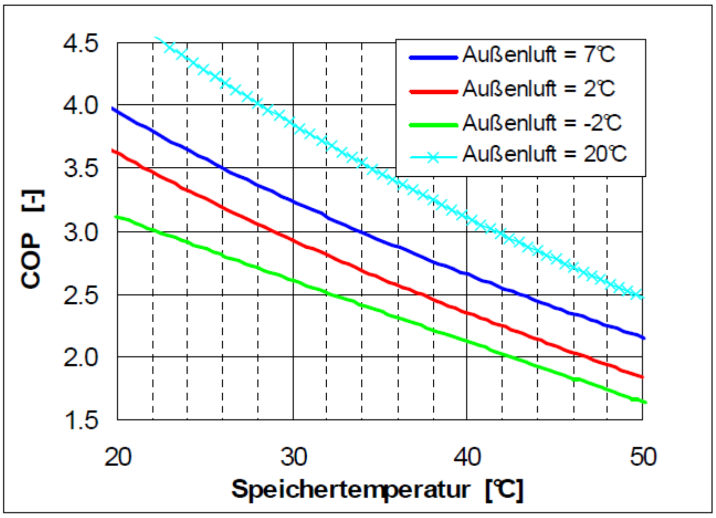


Abbildung 6: Ergebnisse COP Warmwasser

Figure 4—21 DHW COP: Heating from cold to the indicated storage temperatures (Drexel & Weiss)

The DHW COP is the arithmetical mean based on the heating from cold data, graphically illustrated in Figure 4-21 and tabulated in Figure 4-22.

Aus dem Diagramm ergeben sich folgende mittlere $COP_{Aufheiz}$:

Außenlufttemperatur [°C]	$COP_{Aufheiz}$ [ ]	thermische Leistung [kW]
-2°C	2.31	0.84
2°C	2.61	1.00
7°C	2.96	1.16
20°C	3.47	1.49

Figure 4—22 DHW Average COP for outdoor temperature (Drexel & Weiss)

Reheating following tapping is described thus: The cylinder water is run off until the compressor re-starts based on the signal from the cylinder thermostat. The COP is now evaluated for the reheat period resulting with the following temperatures:

Cylinder temperature: 50.1°C

Cylinder temperature at compressor on: 42.4°C

Mean cylinder temperature: 46.2°C

The COP results for *Nachladung* or "reheating" are provided in Figure 4-23. Note the impact of heat loss from the compressor. At low temperatures, the compressor heat loss can provide space heating "*mit Abwarme des Kompressors*" or "with waste heat from the compressor"; the overall COP is increased.

Aus den ermittelten Daten ergibt sich folgendes COP für die Nachladung (bei einer Mitteltemperatur von 46,1°C):

Außenlufttemperatur [°C]	$COP_{Nachladen}$ ohne Abwärme des Kompressors [ ]	thermische Leistung [kW]	$COP_{Nachladen}$ mit Abwärme des Kompressors [ ]	thermische Leistung [kW]
-2°C	1.83	0.78	2.08	0.88
2°C	2.02	0.93	2.39	1.10
7°C	2.33	1.11	2.71	1.28
20°C	2.71	1.41	(2.93)	(1.52)

Figure 4—23 DHW reheat COP (Drexel & Weiss)

It is unclear from the document whether a shunt pump is used during the reheat process although it is likely. The report refers to tapping until the heat pump switches on after running off 10 litres of water. Stratification within the cylinder would hardly lead to the cylinder thermostat picking up a cold water ingress of 10 litres at 20°C in a 200 litre store at 50°C. A mass balance for these temperatures and volumes provides a bulk cylinder temperature of 48°C, some 2K below set point. It would appear that water is drawn off until the bulk temperature reduces to 40°C. Similarly, only pumped mixing would enable the assessment of total cylinder temperature as the water is raised back to set point.

Pilot Study: Field experiments, Daikin Altherma and ClimaCheck

Barratt Developments kindly provided the opportunity between January and June 2010 to assess the domestic hot water performance of an 'as-installed' Daikin Altherma mono-block air source heat pump at the Barratt Green House, Innovation Park, Building Research Establishment, UK. The mono-block is a split unit with an external evaporator unit connected by refrigerant pipework to an internal heat exchanger, the "Hydrobox", which contains the condenser heat exchanger to the central heating system, Figure 4-24. The two units are connected by refrigerant pipework, containing in this case R410A, thus requiring qualified and registered “F Gas” operatives for its installation.

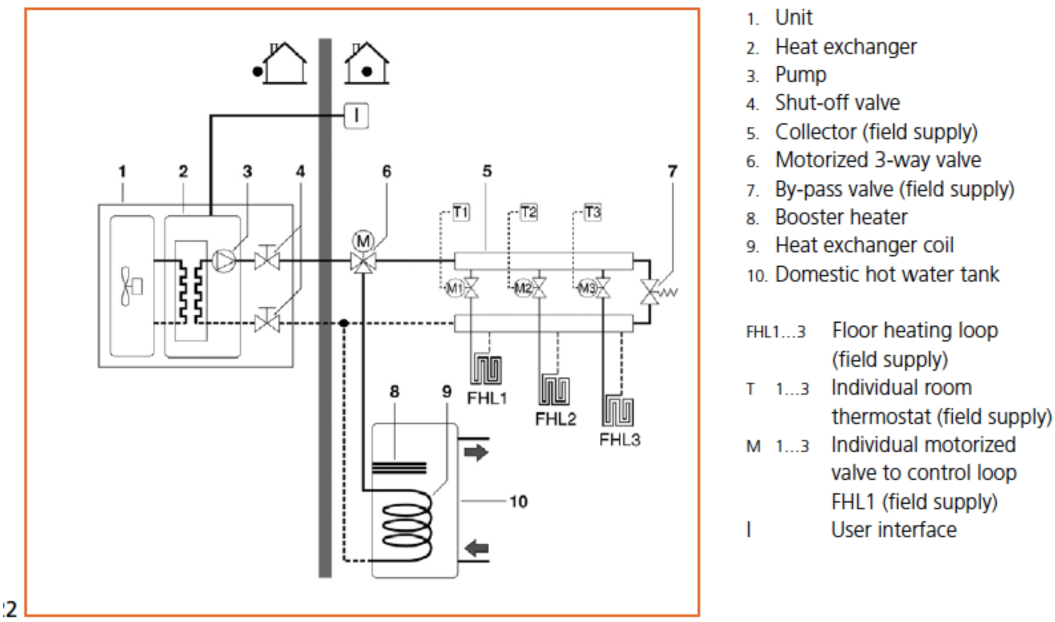


Figure 4—24 Daikin Altherma Bochure (Daikin, PCAWUSE11-06B)

The heat pump was connected to a ClimaCheck[7] monitoring unit designed to measure the operation of the vapour compression cycle. The ClimaCheck has been designed primarily for the air conditioning market to enable the calculation of performance directly from the vapour compression cycle and to provide the coefficient of performance (COP) at one minute intervals. This is a critical difference between the ClimaCheck and the more usual monitoring methods, where performance is calculated from heat meters inserted across the sink flow and return pipes, along with electrical metering of power/energy into the unit. ClimaCheck measurements do not include any heat losses at the condenser heat exchanger, the casing, or those associated with the pipework or parasitic electrical losses. Its role is to assess the operation of the critical components of the vapour compression cycle, the compressor and expansion valve, in order to optimise performance by targeted maintenance. Output from monitoring is downloaded from the home-site for each day of operation, the data output providing refrigerant circuit pressures and temperatures, sub-cooling, super-heating and isentropic compression efficiency along with voltage and current to provide COP.

Previous to the pilot study, space heating performance data was collected for three months of the autumn/winter period during 2009. In January 2010 this researcher was provided access to assess the domestic hot water (DHW) performance. The Barratt Green House is unoccupied so

[7] http://climacheck.com/

DHW performance can be readily assessed for both heating from cold and for tapping re-heat by approximating the EN 255-3:1997 regime based on emptying half the cylinder and monitoring the re-heat process.

Such field experimentation provides valuable insight into the challenges encountered in monitoring, not the least of which were incomplete commissioning of the hot water system including mixed hot water at the bath (the only high flow rate tap in the building) and necessitating long periods in between testing waiting for the 300 litre cylinder to empty and refill, thus restricting the number of test runs possible. More importantly, a 'hands on' approach provides access to raw data from the monitoring system where, for a suitably trained engineer, any anomalies in the recorded performance should become apparent.

Operating data downloaded for the 5 February 2010, where DHW operation was being monitored, proved to be critical for the analysis of all previously collected performance assessment. The download for 14.45 hours showed the following conditions, Table 4-6:

Mid evaporator temp	0°C
Refrigerant low pressure	6.97 bar (g)
Compression in	1.4 °C
Superheat	1.4 °C
Mid condensing	46.7 °C
Expansion valve	40.4 °C
Super cool	6.3 °C
Compressor out	74.4 °C
Compressor Isen' effic'	75.1%
Compressor power	1.87 kW
COP Cool	**3.34**
Capacity Cool (kW)	6.2 kW
SecC flow rate	0.32 kg/s
Power	1.9 kW
current	7.9 amps
voltage	239.4 v

Table 4—6 Climacheck data

It had been assumed that since the software was developed principally to monitor refrigeration systems for cooling, the 'COP Cool' column heading in the downloaded spreadsheet actually referred in this case to 'COP heating' since the system was set up as a heat pump, resulting in a COP_{HP} of 3.4.

The data from Table 4-6 was inserted into a Pressure Enthalpy (Ph) diagram for R410a, Figure 4-25. The approximated COP heat pump value was as follows:

$$\boldsymbol{COP_{HP} = \frac{Qout}{Win} = \frac{(470-270)}{(470-425)} = 4.44}$$

The ClimaCheck should have shown a COP_{HP} somewhere in the region of 4.44.

The calculation of COP refrigeration shows:

$$\boldsymbol{COP_{Ref} = \frac{Qin}{Win} = \frac{(425-270)}{(470-425)} = 3.44}$$

The ClimaCheck value of 3.34 for "COP Cool", shows an approximately 3% difference from COP_{Ref} indicating that "COP Cool" did in fact refer to COP cooling rather than COP heating.

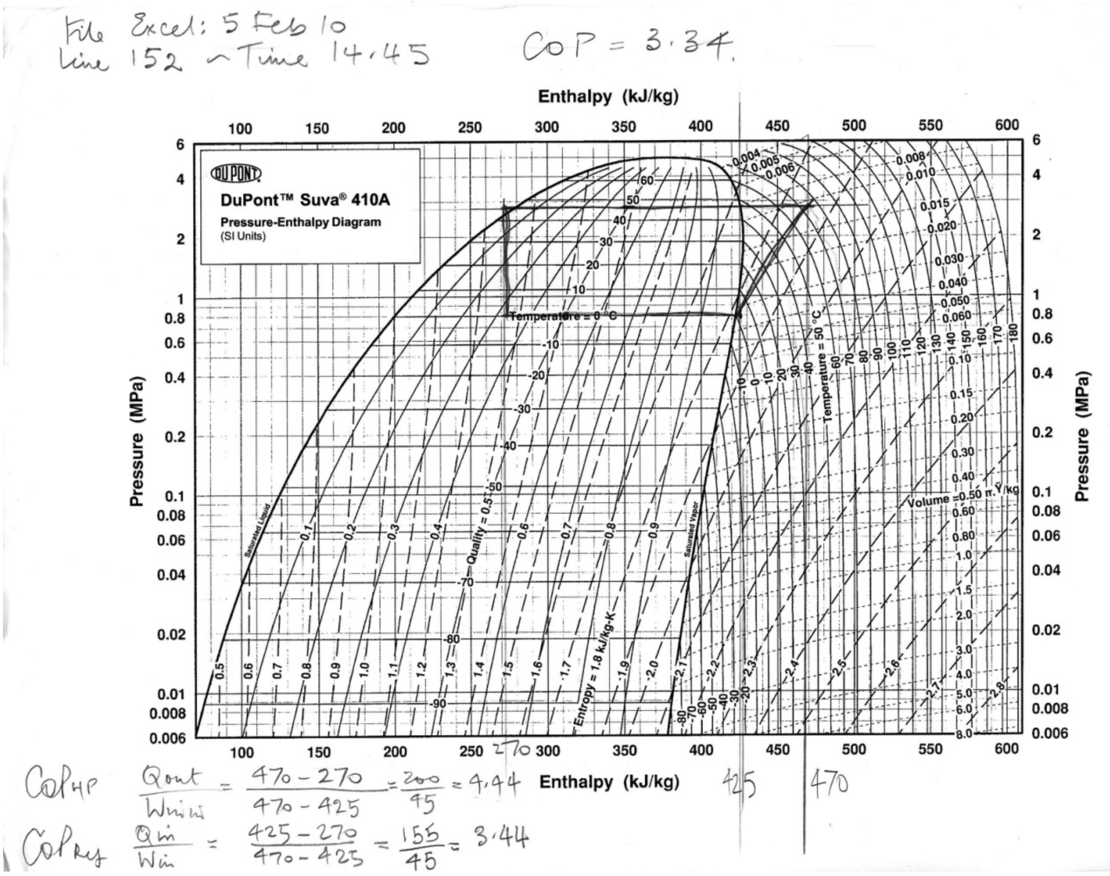


Figure 4—25 Barratt Green House Ph diagram for 14.45 hours 5th Feb 2010

Following this analysis, contact with the ClimaCheck agent confirmed that the software was incorrectly set up and would be subsequently re-set to show COP for heating, COP_{HP}. However, re-setting the data files meant the loss of historic data and therefore it was no longer possible to download previous files in order to correct the data for winter heating performance.

As well as this fault in the calculation of COP_{HP}, analysis of the Altherma heat pump components identified another monitoring anomaly. The Hydrobox contains a resistance heater, “backup heater”, that operates during low ambient temperatures and for de-frost when the heating system temperature is below 18°C, Figure 4-26.

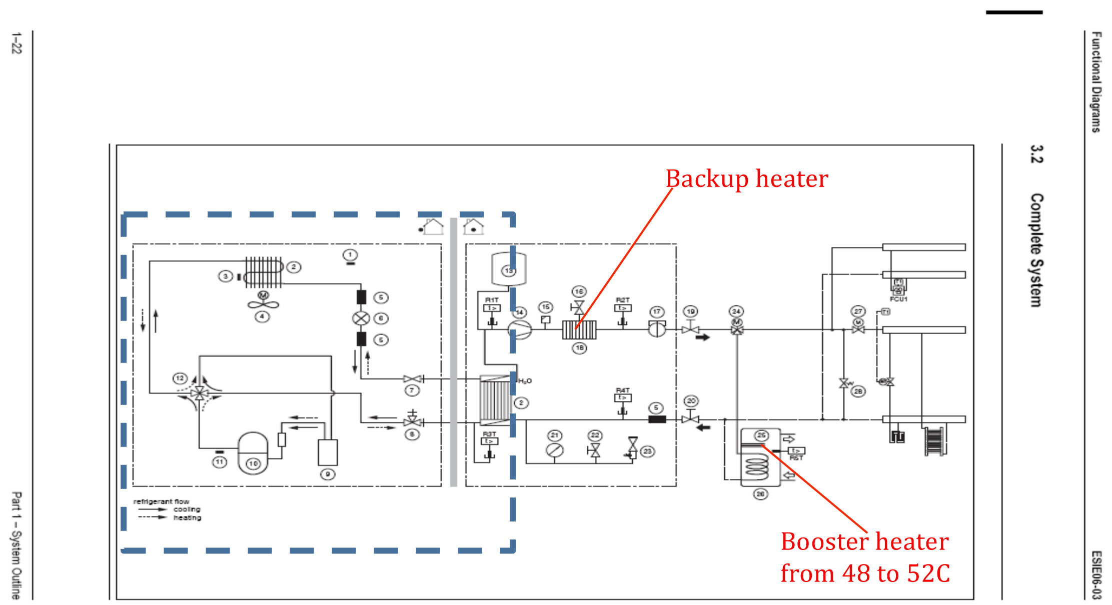


Figure 4—26 Boundary of the ClimaCheck (blue dashed lines) (after Daikin Altherma, EEDEN09-720)

Since the ClimaCheck monitors only the vapour compression cycle, no separate metering of this resistance heater was undertaken, its heat output thus conflated with that of the heat pump output. It was no longer possible to definitively state the winter heating performance since the input to the “backup heater” was not measured, similarly for DHW assessment during cold weather where de-frosting was required. Furthermore, as noted, there is no monitoring of the parasitic electrical energy required to run the Hydrobox controls or of the heating circulation pump.

These examples of incorrect evaluation of monitoring protocol provide a wider resonance applicable to all field trials. It is imperative that the designers of field trials understand the mechanics of each monitored appliance and its installation system and not presume that installers of monitoring equipment will necessarily understand the objectives of the monitoring and thus the correct set-up of the monitoring system.

ClimaCheck and Domestic Hot Water production

The time required to run more than one DHW test per day, allied with a persistent heat pump fault and no access to the building outside of visiting hours, resulted in a limited sample of 'heating from cold' and 'tapping re-heat' results, however, this sample provides a useful assessment of these two DHW heating protocols. The samples provide evidence that, for stratified cylinders, there is little difference in performance between the two conditions.

The tapping re-heat test applied at the Barratt Green House is based on EN 255, from drawing off approximately half the cylinder contents and monitoring from when the heat pump turns on through to its final switching off. The Altherma, with its variable speed compressor[8], is designed to reach approximately 48°C on heat pump power; the cylinder resistance heater or "booster heater", Figure 4-26, may be set to operate for 30 minutes once a week to control legionella. Data gathered during January 2010 for both heating from cold and for tapping re-heat provide COP results of 3.8 and 4.6 that reflect the change in outdoor temperature as ambient climbs from 4.8°C to 16.2°C, Figures 4-27 and 4-28.

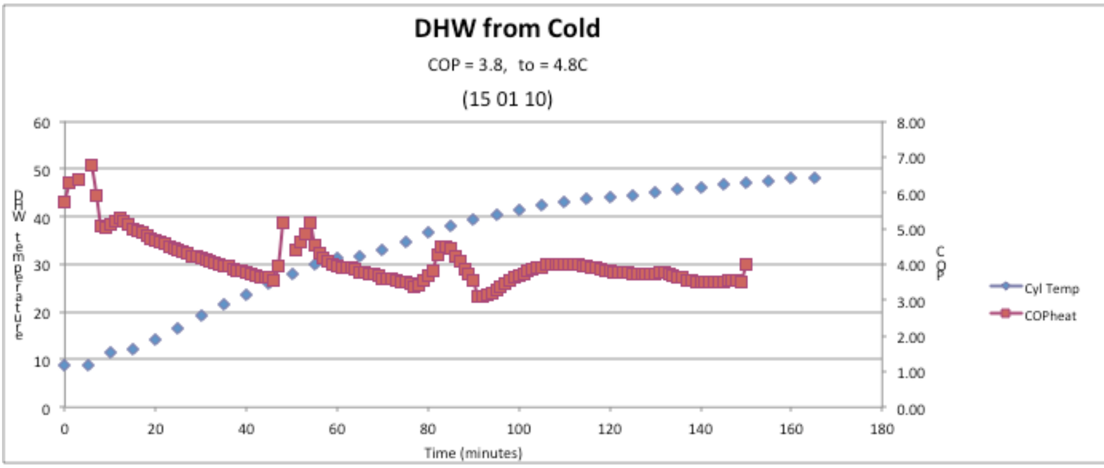


Figure 4—27 DHW heating from cold

[8] Interestingly, the irregular 'blips' in the Altherma COP curves, Figures 4-27 to 4-30, show evidence of the variable speed control readjusting the compressor speed (changing the power-in) to optimise COP during the heating process. The compressor control compensates during operation to maximise efficiency by adjusting speed output. In comparison, the fixed speed compressor COP curves from the BRE (Figure 4-18) and Passivhaus (Figure 4-21) show a much more consistent downward gradient indicating a more linear lowering of COP as DHW temperature rises.

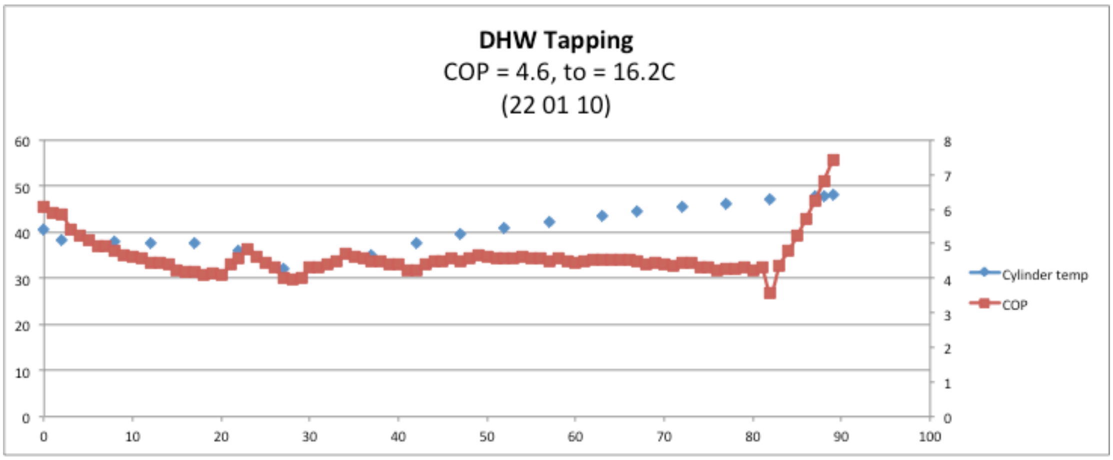


Figure 4—28 DHW tapping re-heat

Comparison with results for both tests from the same day in April (09 04 2010) show that COP from cold is slightly lower than that for tapping although the ambient temperature for the latter condition is slightly higher, Figures 4-29 and 4-30.

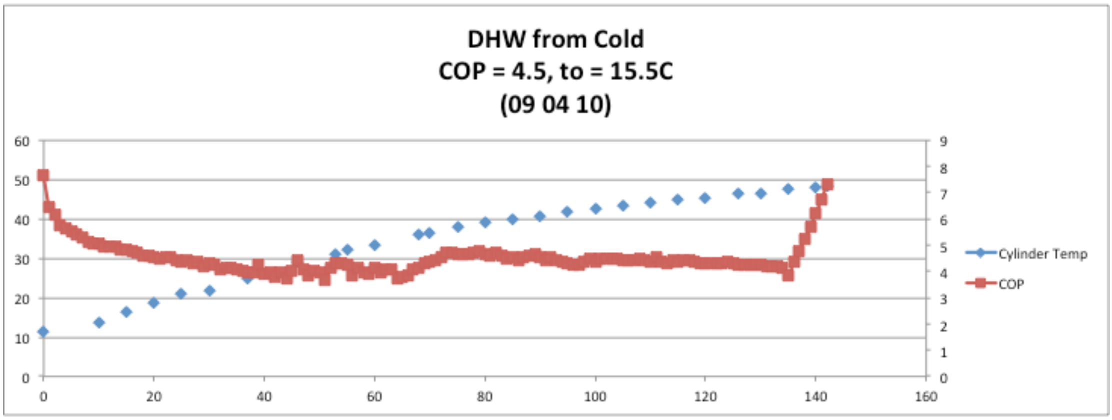


Figure 4—29 DHW heating from cold

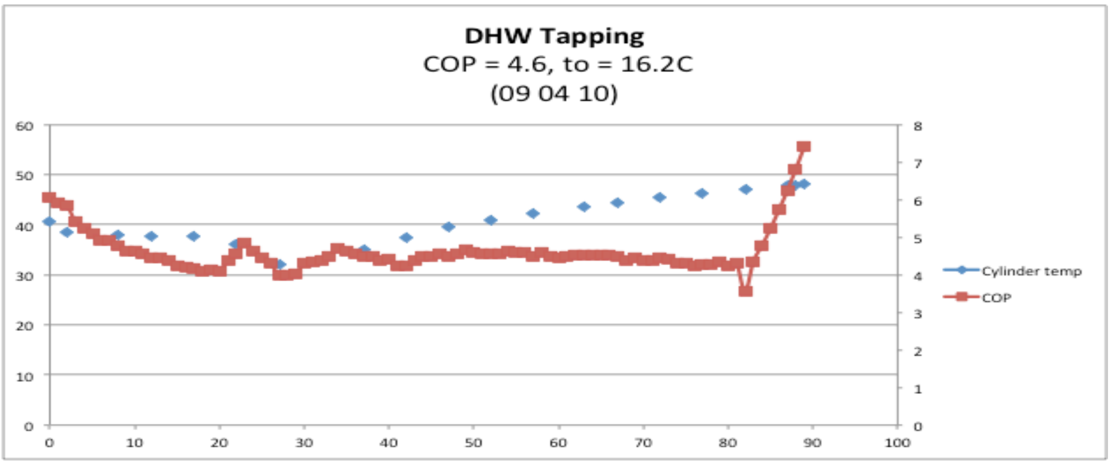


Figure 4—30 DHW tapping re-heat

Plotting all DHW test performances, six from cold and five for tapping, shows little difference between heating from cold and tapping re-heat; COP appears to be solely a function of ambient temperature, Figure 4-31.

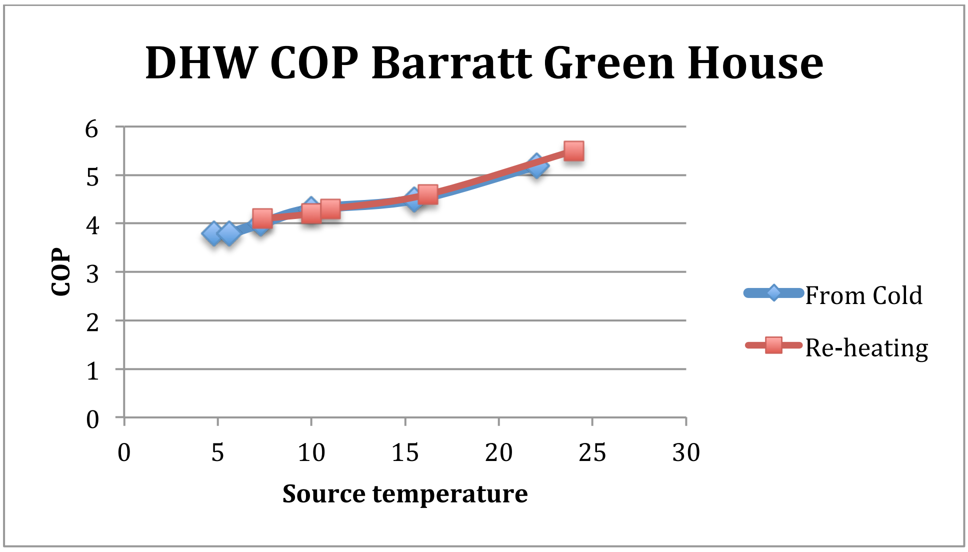


Figure 4—31 DHW, comparison between heating from cold and tapping re-heat

A building specification for the Barratt Green House is provided by the Building Research Establishment (Barratt Developments, undated), where the description of the DHW states:

> "Kingspan Solar Ltd provides solar hot water packages, custom designed for each application. The challenge in the Barratt Green House was to design and install a solar thermal system, in conjunction with other equipment, to achieve a rating of Code level 6. The system was designed, in conjunction with Arup, by Coates Design Partnership and featured a Kingspan Solar system with flat panel collectors and Range Tribune HE Duplex solar cylinder."

The photographs of the installation, Figure 4-32, show a single coil indirect cylinder rather than the "duplex", dual coil described. Solar thermal is integrated through a Daikin supplied externally fixed plate-heat exchanger comprising solar thermal and DHW primary flow and return (the Daikin EKSOLHW). Both of the immersion "booster" heaters shown in the left-hand image are disconnected. The cylinder thermostat, located in the top third of the cylinder just above the higher immersion heater, provides feedback to the heat pump controls of storage temperature. The photograph on the right, Figure 4-32, shows the plate heat exchanger and the relative height of the primary coil flow and return in the bottom third of the cylinder.

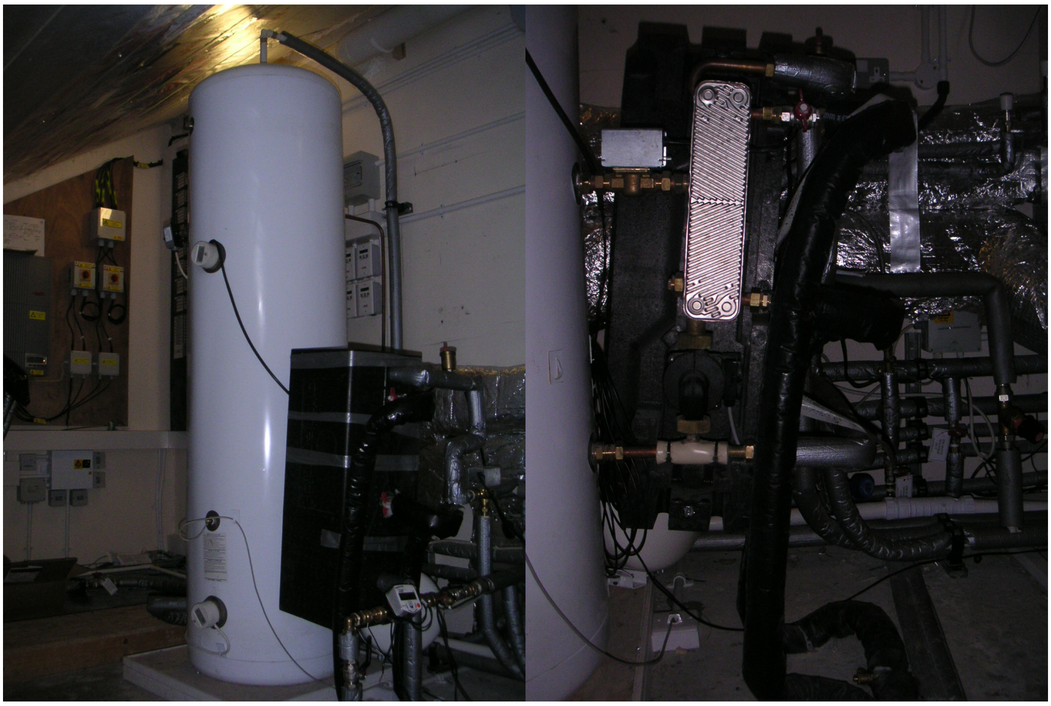


Figure 4—32 Barratt Green House DHW cylinder with Daikin plate heat exchanger

The heat pump switches on when the thermostat reads around 37 or 38°C, a 10K switching differential. Since the cylinder is stratified, by the time the thermostat registers 38°C, the primary coil is immersed in cold feed water. The temperature differential driving heat exchange between the primary flow from the heat pump and the stored water is identical for both heating from cold and for tapping re-heat, hence the similarities between the COPs. The only substantive difference between the two methods is the time taken, tapping from cold takes longer since the whole cylinder must be heated.

Both the Mitsubishi Ecodan and the Daikin Altherma have variable speed compression with electronic expansion valves and the DHW test results should provide some insight into the efficiency of this particular heat pump design. Unfortunately, published efficiency for the Ecodan is from the unit operating at the EN 14511 fixed speeds of 50% and 100% and with a DHW set point temperature of 55°C in comparison to the Altherma's 48°C. Additionally, the BRE's inclusion of pumping power for the condenser (EN 14511 methodology) and pipe and cylinder losses, also contribute to the difference in performance. However, for both the Altherma and the Ecodan, DHW efficiencies are impressive, with COPs ranging from 3 to over 5, Table 4-7.

Manufacturer	7°C	12°C	20°C	25°C
Ecodan (Speed 4)	3.16	3.63	4.25	4.9
Altherma (Variable speed)	4.0	4.4	4.85	5.6

Table 4—7 Comparison of BRE Ecodan DHW results with ClimaCheck Altherma

The ClimaCheck DHW experimental results indicate that for stratified cylinders there is little difference between heating from cold and tapping re-heat due to the immersion of the cylinder coil in cold feed water in both cases. Maximum DHW annual efficiency is therefore more likely to be achieved by heating large volumes of water rather than small.

Seasonal performance and the Bin Method

It is apparent that DHW presents a number of problems for assessing the likely performance in the field, not the least of which is that manufacturers generally fail to provide EN 255-3 or EN 16147 DHW test data and that a test based on EN 14511 is not reflective of the heat transfer process for stored hot water production. However, given that manufacturers do provide EN 14511 results for space heating, this may be the only source data to assess the potential seasonal efficiency of heat pumps in a typical space heating and domestic hot water central heating set up. Assessing potential performance was one the key outputs assigned by the International Energy Agency to the Annex 28 research project. The methodology applied was the “bin method”.

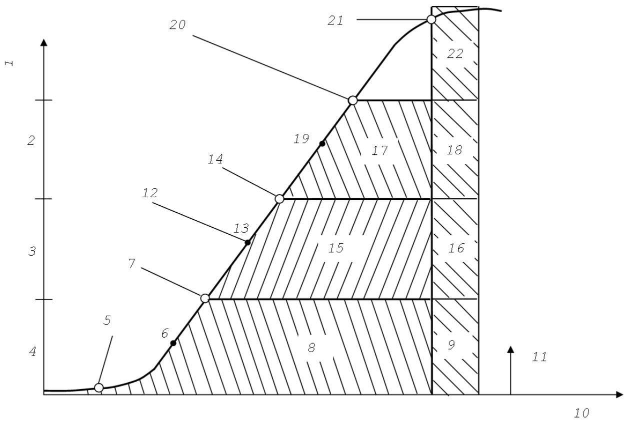
The bin method was used to assess residential exhaust air heat pump seasonal performance for the IEA Annex 28 programme (Wemhoner & Afjai, 2006) and has since been incorporated into EN 15316-4-2: 2008, “Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 4-2: Space heating generation systems, heat pump systems” (BSI, 2008).

Wemhoner and Afjai state:

> “Differences between the calculation and the measured values are in the range of ± 6% for the seasonal performance factor. Considering the simplifications in the approach these values are satisfactory and show the applicability of the method,” pVIII.

With a plus or minus 6% match to monitored data, the bin method would appear to be an appropriate estimation tool for seasonal performance and therefore its use is explored. The method especially suits exhaust air and air source heat pump assessment where source air temperature data for the installation location is available. EN 14511 COP data is required for a range of external temperatures and EN 255-3 COP data for hot water production. This data is then applied to the building heat loss and hot water load. The source temperature is divided into bins centred on the COP test temperatures, Figure 4-33.

EN 15316-4-2:2008 (E)



Key

1 cumulative bin hours [h]
2 t_3
3 t_2
4 t_1
5 design outdoor temperature
6 OP_1
7 $\theta_{1,\mathrm{hlim}} = \theta_{2,\mathrm{llim}}$
8 SH_1
9 DHW_1
10 outdoor air temperature [°C]
11 direction of cumulating temperature difference (space heating load) and DHW load over time
12 θ_2
13 OP_2
14 $\theta_{2,\mathrm{hlim}} = \theta_{3,\mathrm{llim}}$
15 SH_2
16 DHW_2
17 SH_3
18 DHW_3
19 OP_3
20 upper ambient temperature for space heating = $\theta_{3,\mathrm{hlim}}$
21 design indoor temperature
22 DHW_4

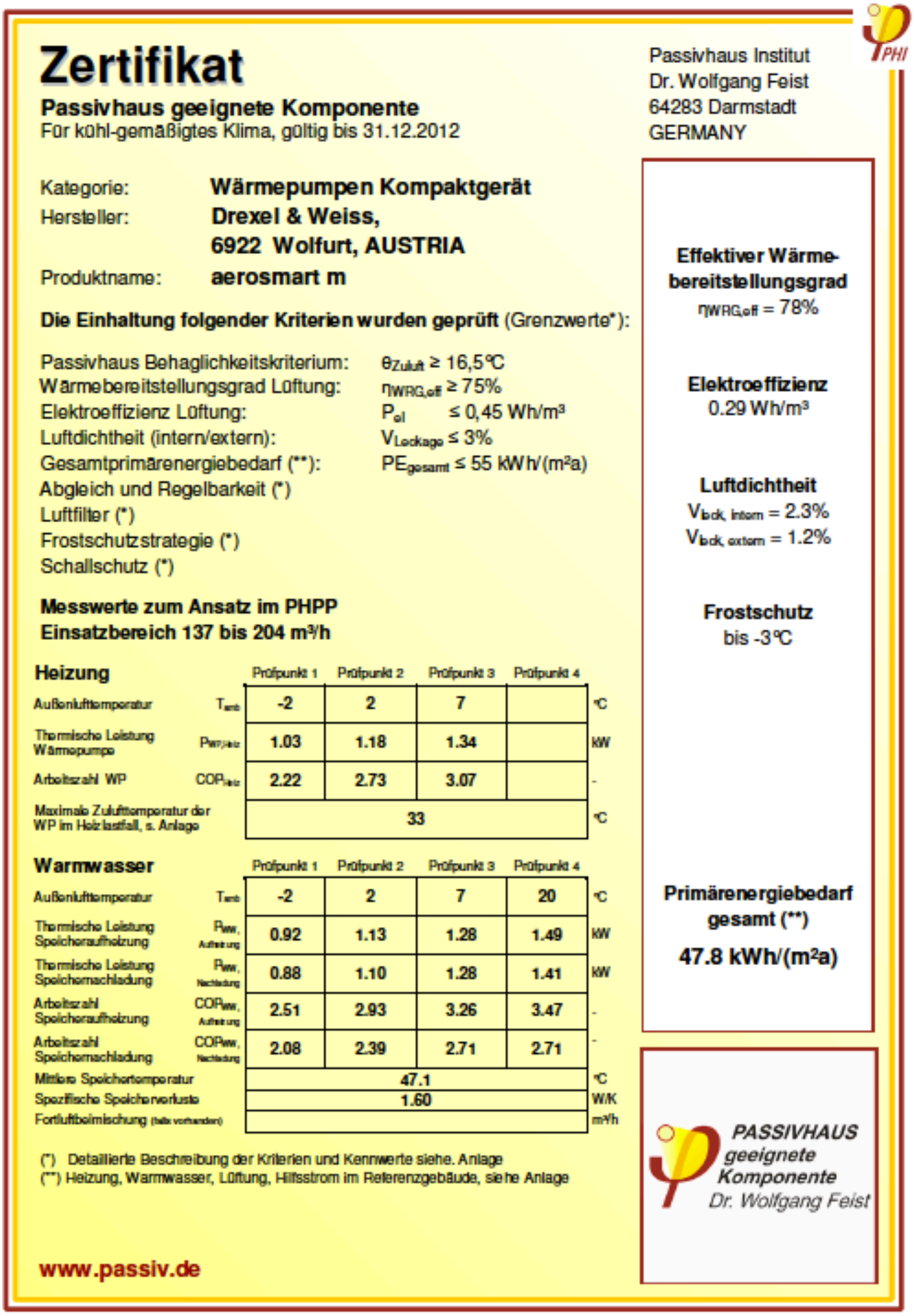
Figure 4 — Bin hours vs outdoor air temperature – sample with 3 bins for space heating (SH) and constant daily domestic hot water (DHW) heat energy requirement (4 bins for DHW)

Figure 4—33 Bin Method (EN 15316)

A heat loss calculation for the building will provide the heat loss coefficient (HLC) in (W/K), the local weather-file provides the temperature and duration for each temperature bin. Thus the annual space heating energy (kWh) is calculated as a function of HLC and bin temperature duration. A similar process provides annual performance for DHW where annual DHW load is divided between the temperature bins based on an annual assessment of hot water demand. Both processes are based on assumptions of COP at set ambient temperatures, assumptions we have already seen fit to question.

Bin method compact unit heat pump example

The application of the bin method in EN 15316-4-2:2008 is dependent on sufficient test data from the heat pump manufacturer for both space heating and DHW performance. The Passivhaus Institute provide such information for heat pump driven combined service units. Drexel and Weiss manufacture such a unit and provide test data supported by quality controlled Passivhaus *Wämepumpenkompaktgeräte Zertifizierung*, heat pump compact unit certification, Figure 4-34.



Zertifikat

Passivhaus geeignete Komponente
Für kühl-gemäßigtes Klima, gültig bis 31.12.2012

Passivhaus Institut
Dr. Wolfgang Feist
64283 Darmstadt
GERMANY

Kategorie: **Wärmepumpen Kompaktgerät**
Hersteller: **Drexel & Weiss,**
6922 Wolfurt, AUSTRIA
Produktname: **aerosmart m**

Die Einhaltung folgender Kriterien wurden geprüft (Grenzwerte*):

Passivhaus Behaglichkeitskriterium: $\theta_{Zuluft} \geq 16{,}5°C$
Wärmebereitstellungsgrad Lüftung: $\eta_{WRG,eff} \geq 75\%$
Elektroeffizienz Lüftung: $P_{el} \leq 0{,}45\ Wh/m^3$
Luftdichtheit (intern/extern): $V_{Leckage} \leq 3\%$
Gesamtprimärenergiebedarf (**): $PE_{gesamt} \leq 55\ kWh/(m^2a)$
Abgleich und Regelbarkeit (*)
Luftfilter (*)
Frostschutzstrategie (*)
Schallschutz (*)

Effektiver Wärmebereitstellungsgrad
$\eta_{WRG,eff} = 78\%$

Elektroeffizienz
$0.29\ Wh/m^3$

Luftdichtheit
$V_{lock,intern} = 2.3\%$
$V_{lock,extern} = 1.2\%$

Frostschutz
bis -3°C

Primärenergiebedarf gesamt (**)
47.8 kWh/(m²a)

Messwerte zum Ansatz im PHPP
Einsatzbereich 137 bis 204 m³/h

Heizung

		Prüfpunkt 1	Prüfpunkt 2	Prüfpunkt 3	Prüfpunkt 4	
Außenlufttemperatur	T_{amb}	-2	2	7		°C
Thermische Leistung Wärmepumpe	$P_{WP,Heiz}$	1.03	1.18	1.34		kW
Arbeitszahl WP	COP_{Heiz}	2.22	2.73	3.07		-
Maximale Zulufttemperatur der WP im Heizlastfall, s. Anlage		33				°C

Warmwasser

		Prüfpunkt 1	Prüfpunkt 2	Prüfpunkt 3	Prüfpunkt 4	
Außenlufttemperatur	T_{amb}	-2	2	7	20	°C
Thermische Leistung Speicheraufheizung	$P_{WW,Aufheizung}$	0.92	1.13	1.28	1.49	kW
Thermische Leistung Speichernachladung	$P_{WW,Nachladung}$	0.88	1.10	1.28	1.41	kW
Arbeitszahl Speicheraufheizung	$COP_{WW,Aufheizung}$	2.51	2.93	3.26	3.47	-
Arbeitszahl Speichernachladung	$COP_{WW,Nachladung}$	2.08	2.39	2.71	2.71	-
Mittlere Speichertemperatur		47.1				°C
Spezifische Speicherverluste		1.60				W/K
Fortluftbeimischung (falls vorhanden)						m³/h

(*) Detaillierte Beschreibung der Kriterien und Kennwerte siehe. Anlage
(**) Heizung, Warmwasser, Lüftung, Hilfsstrom im Referenzgebäude, siehe Anlage

PASSIVHAUS geeignete Komponente Dr. Wolfgang Feist

www.passiv.de

Figure 4—34 Drexel & Weiss Aerosmart Combined Service Unit heat pump test certificate (Passivhaus)

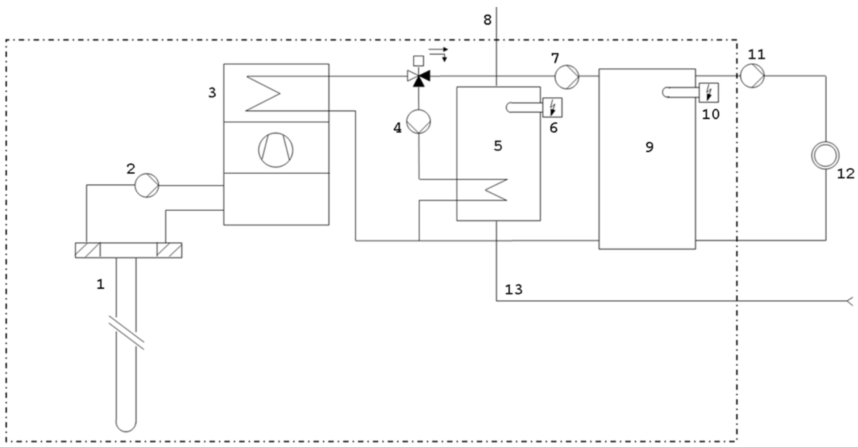
The Passivhaus methodology requires space heating source temperature testing, *Heizung*, at three temperatures, (-2), 2 and 7°C. Note that the minimum space heating COPs from Passivhaus start at (-2)°C, irrespective of lower ambient temperatures and that the maximum supply air temperature, *zulufttemperatur* is, for this particular model, 33°C. The 13K temperature difference between supply and comfort temperature at 20°C is required to offset the minimal heat losses associated with Passivhaus envelopes at the design outside temperature at continuous ventilation flow rates. The continuous mechanical ventilation acts in winter as warm air heating system.

The domestic hot water, *Warmwasser*, is treated in two ways: heating from cold, *speicheraufheizung*, that is heating from 20 to 50°C, and reheating, *speichernachladung*, from 40 to 50°C. Note also that there is no single overall COP, SCOP or SPF value for the unit since this will be dependent on local climate region in which it is installed; calculating its value is the objective of EN 15316.

To explore the bin method using Drexel and Weiss test data, a 3 bedroom low energy detached house of 96m^2 is modelled in UK Standard Assessment Procedure software (SAP 2006). The EN 15316 methodology should be based on the suite of EN 12831:2003 standards: "Heating systems in buildings — Method for calculation of the design heat load". The UK SAP procedure, whilst not a heat loss calculation, does provide a short cut for presenting the EN 15316 methodology since a SAP worksheet for the building produces the following:

- heat loss coefficient 77 W/K, (heat loss parameter 0.808 W/m^{2}K),
- space heating load of 5314 kWh
- domestic hot water load of 3279 kWh.

Although SAP uses degree-day information from the East Pennines, the following analysis is based on the readily available CIBSE Heathrow outdoor weather data. As can be seen from EN 15316-4-2, Figure 4-35, SAP provides a reasonably analogous comparison since cylinder losses are included within the boundary as is pump energy.



Key

1 heat source system (here: vertical borehole heat exchanger)
2 source pump
3 heat pump
4 DHW storage loading pump
5 DHW storage
6 DHW back-up heater
7 primary pump
8 DHW hot water outlet
9 heating buffer storage
10 space heating back-up heater
11 circulation pump space heating distribution subsystem
12 heat emission subsystem
13 DHW cold water inlet

Figure 1 — System boundary of the generation subsystem

Figure 4—35 System boundary (EN 15316)

The bin method example applies the following steps, the area-weighted equations are based on the original work of Wemhoner and Afjai (2006):

1) From the SAP calculations find the heat loss coefficient and plot the heat loss to ambient slope. From SAP internal gains find the balance temperature, Figure 4-36.
2) From weather data, calculate the number of hours per year in each of the bins, Table 4-8.
3) From SAP heat loss coefficient, calculate the space heating power (kW) and energy (kWh) in each bin, Table 4-9.
4) Calculate the cumulative frequency (cusum) for the space heating, Figure 4-37.
5) Divide the cumulative frequency into temperature bands based on manufacturer's test data. This can be problematic since the bins need to be logically divided but some bins are smaller than others, some temperatures are non-standard. In the example, where mean bin temperatures are used, the bin areas are divided to allow COP values for (-2), 2 and 7°C to be representative of the bins, Figure 4-38.
6) Sum for each bin the total kWh. Each bin represents a fraction of the annual space heating requirement, Table 4-10.
7) Find the area-weighted space heating SCOP (Equation A2, Wemhoner and Afjai, 2006 p14), Table 4-11

8) DHW is based on the same approach but using (-2), 2, 7 and 20°C. The annual DHW load is from SAP.

9) Find the area-weighted DHW SCOP (Equation A.6, Wemhoner and Afjai, 2006 p15), Tables 4-12 and 4-13. Note that DHW COP depends on how it is evaluated (from cold or reheat).

10) Calculate the area-weighted SCOP (SPF) for the heat pump, Tables 4-14 and 4-15.

1) Balance temperature

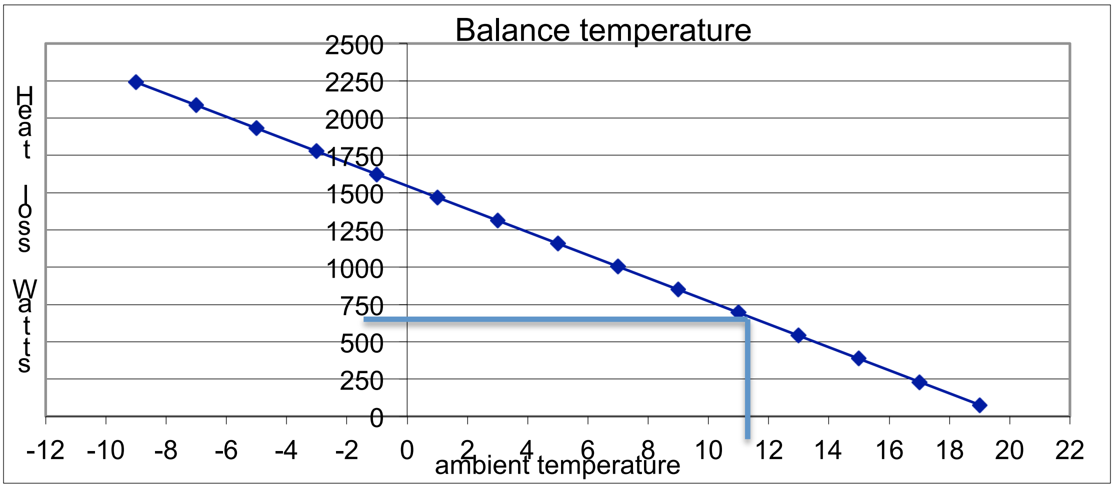


Figure 4—36 Balance temperature of 11°C at 700 Watts heat gains

2) Number of heating hours in each bin based centred on test data

A	B	C	D	E	F	G	H
range	mean to	%	%/100	h/day	h/day at to	d/yr	hours/yr at to
(-9.9) - (-8)	-9	0.02	0.0002	24	0.0048	365	2
(-7.9) - (-6)	-7	0.06	0.0006	24	0.0144	365	5
(-5.9) - (-4)	-5	0.27	0.0027	24	0.0648	365	24
(-3.9) - (-2)	-3	0.85	0.0085	24	0.204	365	74
(-1.9) - 0	-1	2.03	0.0203	24	0.4872	365	178
0.1 - 2	1	4.44	0.0444	24	1.0656	365	389
2.1 - 4	3	6.86	0.0686	24	1.6464	365	601
4.1 - 6	5	8.73	0.0873	24	2.0952	365	765
6.1 - 8	7	10.52	0.1052	24	2.5248	365	922
8.1 - 10	9	12.26	0.1226	24	2.9424	365	1074
10.1 - 12	11	12.28	0.1228	24	2.9472	365	1076
12.1 - 14	13	11.45	0.1145	24	2.748	365	1003
14.1 - 16	15	10	0.1	24	2.4	365	876
16.1 - 18	17	7.82	0.0782	24	1.8768	365	685
18.1 - 20	19	5.25	0.0525	24	1.26	365	460
20.1 - 22	21	3.14	0.0314	24	0.7536	365	275
22.1 - 24	23	1.92	0.0192	24	0.4608	365	168
24.1 - 26	25	1.17	0.0117	24	0.2808	365	102
26.1 - 28	27	0.55	0.0055	24	0.132	365	48
28.1 - 30	29	0.23	0.0023	24	0.0552	365	20
30.1 - 32	31	0.1	0.001	24	0.024	365	9
32.1 - 34	33	0.04	0.0004	24	0.0096	365	4
							8759

Table 4—8 Calculation of bin hours

3)

B	H	I	J	K	L	M
mean to	hours/yr at	kW/K	ta	dT	kW	kWh/yr
-9	2	0.0773	20	29	2.241	4
-7	5	0.0773	20	27	2.086	11
-5	24	0.0773	20	25	1.932	46
-3	74	0.0773	20	23	1.777	132
-1	178	0.0773	20	21	1.622	289
1	389	0.0773	20	19	1.468	571
3	601	0.0773	20	17	1.313	789
5	765	0.0773	20	15	1.159	886
7	922	0.0773	20	13	1.004	926
9	1074	0.0773	20	11	0.850	913
11	1076	0.0773	20	9	0.695	748

Table 4—9 Calculation of space heating power and energy

4) and 5)

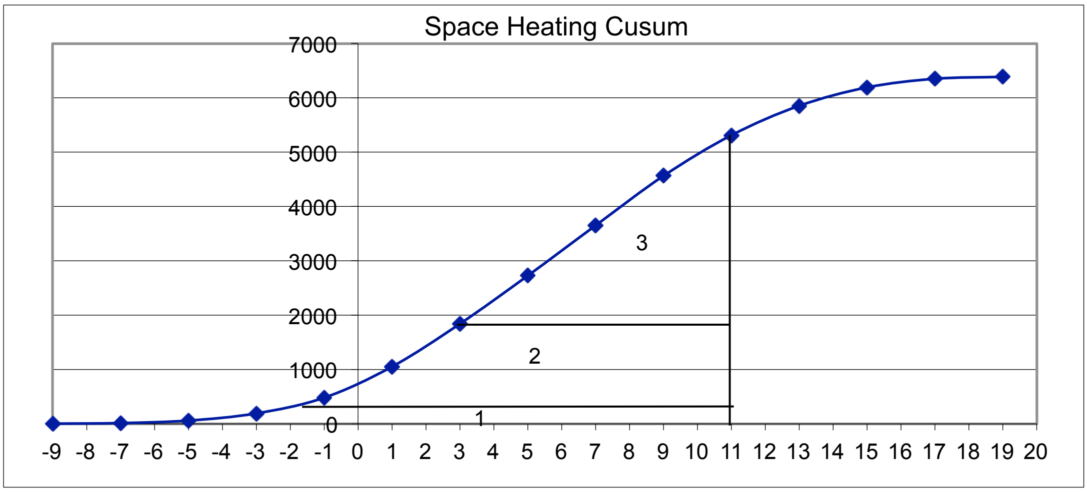


Figure 4—37 Graphical space heating cusum calculation

6)

B	N
mean to	Cusum
-9	4
-7	15
-5	61
-3	193
-1	481
1	1052
3	1842
5	2728
7	3653
9	4566
11	5314

Table 4—10 Tabular space heating cusum calculation

7) Area Weighted space heating SCOP applying Equation A2.

SCOP SPACE HEATING				
	Bin 1	Bin 2	Bin 3	Total

Operating points	-2	2	7	
kWh	481	1360	3473	5314
Bin weighting	0.091	0.256	0.654	1
COP	2.22	2.73	3.07	
COP*bin fraction	0.20	0.70	2.01	**2.91**

Table 4—11 Space heating SCOP

9) DHW weighted COP for re-heat applying Equation A.6.

SCOP DHW tapping re-heat					
	Bin 1	Bin 2	Bin 3	Bin 4	Total
Operating points	-2	2	7	20	
hours	283	990	3836	3650	8759
Bin weighting	0.032	0.113	0.438	0.416	1.000
COP	2.08	2.39	2.71	2.71	
COP*bin fraction	0.07	0.27	1.19	1.13	**2.65**

Table 4—12 DHW (tapping) SCOP

9) DHW weighted COP for heating from cold

SCOP DHW from cold					
	Bin 1	Bin 2	Bin 3	Bin 4	Total
Operating points	-2	2	7	20	
hours	283	990	3836	3650	8759
Bin weighting	0.032	0.113	0.438	0.416	1.000
COP	2.51	2.93	3.26	3.47	
COP*bin fraction	0.08	0.33	1.43	1.45	**3.29**

Table 4—13 DHW (from cold) SCOP

10) Weighted SPF based on DHW re-heat and heating from cold

Heat Pump SCOP (DHW tapping re-heat)			
	SH	DHW	TOTAL
Annual load (kWh)	5314	3279	8593
COP	2.91	2.65	
Fraction	0.618	0.382	
COP*fraction	1.80	1.01	**2.81**

Table 4—14 Annual SCOP (DHW tapping)

Heat Pump SCOP (DHW from cold)			
	SH	DHW	TOTAL
Annual load (kWh)	5314	3279	8593
COP	2.91	3.29	
Fraction	0.618	0.382	
COP*fraction	1.80	1.26	**3.06**

Table 4—15 Annual SCOP (DHW from cold)

Tables 4-14 and 4-15 provide SPF values lying between 2.8 and 3.1 depending on the DHW heating mode. The resulting outputs provide an approximate assessment of what is described in EN 15316 as seasonal performance factor but which is really an assessment of SCOP based on manipulating COP values and therefore is not directly related to real world performance. EN 15316 does include corrections for part load (where results are available) and for the impact of bimodal operation, however, what is evident is the difference in SCOP caused by the hot water heating regime. The efficiency of heat pumps in delivering DHW is critical to achieving low emissions in low energy buildings and is clearly related to the cylinder heat up regime. Unresolved is the difference between pumped secondary water, the shunt pump featured in both the BRE and Passivhaus test regimes, and what was witnessed at the Barratt Green House, that tapping re-heat could resemble heating from cold. Forced circulation from the shunt pump increases heat transfer just as fan-assisted radiators provide more output than those relying natural convection; forced convection must therefore provide a higher COP for DHW.

In a Passivhaus dwelling, DHW energy demand has been assessed as roughly twice that of space heating (Clarke, et al, 2009). Space heating supply air temperature peaks at around 33°C for the compact unit (Figure 4-34), with a lower average over the heating season. DHW temperatures are up to 50°C with, in theory, energy demand depending on the proportion of heating from 20°C and top-up. So DHW is weighted by a significantly lower COP than space heat in the computation of seasonal mean COP or SCOP.

Similar arguments can be deployed for ordinary, non-super insulated dwellings, but COP in such dwellings will in practice be dominated by space heat. The Passivhaus combined unit example and the following air source heat pump example are based on a low energy envelope with a heat loss parameter (HLP) of 0.8 W/m^{2}K. At these conditions, equivalent to Passivhaus fabric energy losses, the DHW as a proportion of the annual load is significantly larger than for most UK retrofit dwellings with their higher HLP, a situation that should change as a result of the EPBD.

Bin method Air to Water Heat Pump example

Applying the bin method to non-compact air to water heat pumps is more problematic. Whilst the procedure appears relatively simple, its application is dependent on the heat pump manufacturer supplying sufficient EN 14511 COP operating data for the various bins. A review of manufacturers' data sheets shows, not surprisingly, that many manufacturers provide only the COP values that provide the highest COP for space heating, generally at 7/6°C (dry and wet bulb) against 30/35°C (flow and return). In addition, few manufacturers provide DHW COP results based on EN 255-3 or EN 16147 and fewer still provide COPs for higher summertime temperatures applicable for assessing domestic hot water only. The following example exemplifies these issues. In order to assess the potential efficiency of air source heat pumps, test data bulletins published by WPZ *Warmepumpen-Testzentrum*, Switzerland (WPZ, 2009) provide sufficient data to attempt a bin analysis. Six air source heat pumps are compared at 35 and 55°C to assess the mean COP values for modelling underfloor heating with DHW using EN 14511 test results. The mean values of the six heat pumps are then applied to the bin method using the same SAP model and weather data as in the Compact Unit example although, it must be noted, that the DHW heating regime is that for a space heating emitter, Tables 4-16 to 4-19.

Make	-7/35	2/35	7/35	-7/55	7/55	20/55
Alpha	2.7	3.5	4.1	1.9	2.7	3.2
Daikin	2.7	3.4	4.5	1.5	2.6	3.3
Harreither	2.4	3.1	3.5	1.8	2.4	3.3
Hautec	2.8	3.3	3.7	2	2.7	3.2
Vaillant	2.7	3.4	4.1	1.9	2.7	3.5
WPM	3.1	3.5	4.3	2	2.9	3.4
MEAN	**2.7**	**3.4**	**4.0**	**1.85**	**2.7**	**3.3**

Table 4—16 Mean COP for ASHP, source WPZ

SCOP SPACE HEATING				
	Bin 1	Bin 2	Bin 3	Total
Operating points	-7	2	7	
kWh	61	1781	3473	5314
Bin weighting	0.011	0.335	0.653	1
COP	2.7	3.4	4	
COP*bin fraction	0.03	1.14	2.61	**3.78**

Table 4—17 Space heating SCOP

SCOP DHW				
	Bin 1	Bin 2	Bin 3	Total
Operating points	-7	7	20	
hours	105	6007	2647	8759
Bin weighting	0.012	0.686	0.302	1.000
COP	1.85	2.7	3.3	
COP*bin fraction	0.02	1.85	1.00	**2.87**

Table 4—18 DHW SCOP

Heat Pump SCOP			
	SH	DHW	TOTAL
Annual load (kWh)	5314	3279	8593
COP	3.78	2.87	
Fraction	0.618	0.382	
COP*fraction	2.34	1.10	**3.44**

Table 4—19 Annual SCOP

Table 4-19 shows an air to water SCOP value of 3.44. This is an average value for an average air source heat pump and should provide a rule of thumb assessment for this type of heat source. The most obvious problem with accepting this value has been outlined in Chapter 1 where EST field trial data provides COP values for air source heat pumps ranging from 1.2 to 3.3. EN 15316 assumes a perfect installation, properly designed and operated reflecting laboratory test conditions. Clearly the bin method result is also influenced by both DHW heating method and primary water modelled at 55°C, chosen because it is the nearest equivalent to primary flow temperatures capable of producing DHW at 50°C. However, as we have stated, a steady state temperature drop from 55 to 50°C, as envisioned by EN 14511, does not reflect the heat transfer characteristics of a thermal store where heat transfer reduces as the temperature difference between primary coil and secondary storage merge, nor, it must be said, of the space heating emitters. This reduction in temperature difference could result in cycling, that is unnecessary losses as the heat pump, unable to dissipate its full load, switches off before meeting the sink set point temperature and resulting in rapid on/off/on/off operation or "hunting" as it is sometimes called.

It is worth quoting extensively the BRE (Grigg & McCall, 1988) on the effects of heat pump cycling and its negative impact on performance:

> "Each time the heat pump compressor is stopped the pressure difference between the evaporator and condenser is, at least partially, maintained by the valve gear of the compressor and the action of the thermostatic expansion valve. Restarting the compressor in this state would impose a high load, resulting in an undesirably high starting current, and reduce the service life of the compressor. Compressor loading is therefore normally reduced by opening the refrigerant circuit from condenser to evaporator for a brief period before restarting. During this period liquid refrigerant boils out of the condenser and migrates to the evaporator, cooling the condenser and warming the evaporator……Switching losses would not be significant at the cycling frequencies normal for space heating systems….but would become so during house prewarming at moderate ambient temperatures and particularly when heating domestic hot water. At these times a demand thermostat may be calling for heat to be continuously supplied. However the radiator circuit will be unable to dissipate the continuous full load output power of the heat pump."

With regard to ground source heat pumps, no data is readily available on typical ground temperatures for all locations at varying depths. Ground temperature is subject to change during the year as a result of the heat balance between solar insolation and heat extraction and complicated by ground loop length, depth and the dynamics of ground watertable on soil conductivity. Whilst the bin method "may" suit air source heat pumps, there is simply not enough available data on ground heat transfer conditions for its application to ground source heat pumps although EN 15316 does provide such an example.

The bin method provides a value for SCOP, which claims to be an assessment of SPF, but in which there is limited confidence due mostly to the lack of appropriate EN 14511 or EN 16147 manufacturers' test data and their relevance to real world conditions. The model does not consider part load operation, although EN 15316 does allow for part load operation where these results are available. The analysis implies that only field trial monitoring will provide a reliable estimate of what can be expected in terms of heat pump SPF.

System Boundaries

In order to define system efficiency such as SCOP or SPF there needs to be a system boundary. A heating system may be defined as a number of systems within ever expanding boundaries emanating from the heat source to the dwelling envelope or, indeed, the electrical power grid and beyond. The aim of system boundary definition is to enable like-for-like comparison of efficiency between the same types of plant or for comparison between different heat sources; thus its clear definition is imperative. The EN 15316 bin method described above considers the system efficiency based on laboratory test conditions from EN 14511-3. The system is just the heat pump attached to a load designed to extract maximum heat irrespective of any shortcomings in the source or sink design hydraulics, back up or controls.

We need to define a common boundary for all heat pump tests in order to design a universal monitoring protocol. EN 14511, EN 255-3, EN 16147, the BRE and Passivhaus test methodologies all consider the boundary to include just the heat pump with measurement of flow rate, temperature rise and pump resistance through the heat pump components. All other losses such as cylinder heat loss, sink pipework heat losses and pressure drops are subtracted from the denominator for COP calculation. Only the BRE Mitsubishi DHW report includes cylinder losses but it does state that the test is specifically for the particular Glynwed cylinder.

Comparisons of field trial data are beset with challenges in determining the boundaries for cross comparison. Monitoring protocols must be based on a common approach to boundary definition. Monitoring the EN 14511 procedure introduces complexities associated with logging mass flow rate, temperature difference between flow and return, pressure drop through heat exchangers (ΔP_{4-3} and ΔP_{2-3}) and metering compressor, pumps and/or fans, Figure 4-38. The simplest set up to achieve an EN 14511 equivalent is, in practice, far from simple to install, data log and interpret.

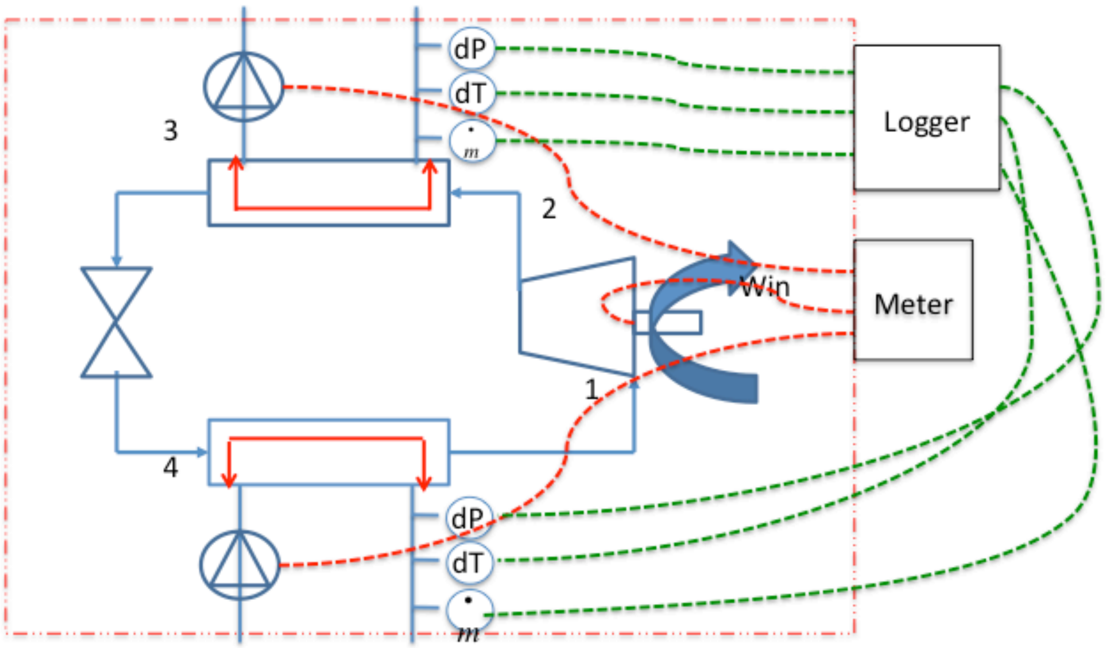


Figure 4—38 Logging and metering, minimum requirements

Summary

We have identified a number of anomalies associated with laboratory testing of heat pumps and the modelling of seasonal performance that will impact on field trial monitoring. Primarily, manufacturers are inconsistent with the data they provide and many publish only the minimum test requirements for EN 14511. Even the MCS 'Product certification requirements document', Issue 2.1 (MCS, 2009) focuses solely on a minimum COP at a single "standard rating conditions". This makes it far from easy to apply the EN 15316-4-2: 2008 bin method at the design stage in order to assess the potential SPF through SCOP for comparison with alternative forms of heating for an emissions analysis.

EN 14511 provides the COP for the heat pump only, its efficiency in converting heat and work into useful heat output. The test rig is designed to extract the maximum output at set source and sink temperatures. Sink conditions are constant with fixed flow and return temperatures. Real systems have a dynamic load ranging from maximum output at design temperature to minimum at balance point.

EN 14511 is not designed to model the dynamic changes associated with field installation. Where load does not match output, either due to insufficient heat transfer surface or insufficient temperature difference, from for example the primary coil to cylinder, the heat pump will switch off on its internal thermostat leading to lower COPs as the heat pump restarts on each cycle.

Whenever the heat pump switches off, residual heat is lost from the unit. For heat pumps installed externally, this heat is dissipated to the atmosphere, for those installed internally, the residual heat is potentially useful in winter but wasteful in summer in overheating the heat pump enclosure.

Testing assumes that source temperature and humidity are constant. For air source heat pumps high humidity will increase heat transfer due to wetting and thus latent heat transfer. For the same rate of heat transfer, the temperature drop through the air source evaporator will be lower for humid air but higher for dry. At very low temperatures, absolute humidity is low, leading to low levels of frosting.

EN 14511 tests for inverter driven pumps are made at 100% and 50% output. These compressor speeds are held constant during the test. In practice, inverters respond during loading to the instantaneous power demand, resetting pump speed to achieve more efficient operation.

DHW presents a number of problems for assessing the likely performance in the field, not the least of which is that manufacturers fail to provide EN 255-3 or EN 16147 even when domestic heat pumps generally supply both space heating and hot water. EN 14511 testing is not reflective of the heat transfer process for stored hot water production. For all the limitations associated with a COP based on set tapping regimes and also with no reference to heating from cold, EN 16147 does provide some indication of COP for DHW. The Passivhaus methodology clearly shows a difference in COP between heating from cold and reheating. With a sufficiently large and insulated cylinder, where stratification is achieved through baffling the cold feed, a DHW regime operating on a single reheat from relative cold should provide a higher COP than constant topping up, but possibly at a cost of higher heat losses from a large hot water store.

EN 14511 implies that ground and water source heat pumps will be housed in an ambient space between 15 and 30°C with sufficiently low relative humidity to prevent dew point wetting and subsequent freezing; this is certainly not always the case with, for example, manufacturers such as Calorex supplying weather proof cabinets for externally housed ground source heat pumps, thus providing the potential not only for frosting due to high ambient humidity at low air temperature but also heat losses from the compressor and from sink pipework. Heat pump fans and compressors are noisy and it is not uncommon for ground source units to be installed in an unheated “boiler house” or for air source units to be installed some distance away from the building.

EN 14511 considers only the pump power demands to overcome resistance in the heat exchangers. In practice pump energy is converted to heat (pump motors are designed to be water cooled) which will fractionally raise the sink water temperature and where pumps are installed internally, all their pumping energy is converted to heat thus providing a useful increase in winter space temperature.

Finally, the underlying assumption of the bin method is that the system is designed, installed and controlled for optimum operation. The central heating system comprises a “correctly” sized ground loop (where applicable), a “correctly” sized heat pump, “correctly” designed space heating and hot water cylinder and occupants who understand the controls and their optimisation.

Since establishing the seasonal efficiency of heat pumps is the ultimate aim of this research, it is necessary to provide a robust monitoring protocol that reflects both EN 14511 and the additional system back up heaters assessed in EN 15316. Real systems require circulation pumps and controls, with many providing back up such as immersion heaters, all of which are integral parts of the legitimate load if whole system efficiency is to be assessed. Establishing appropriate system boundaries and monitoring protocols is therefore the subject of the next chapter.

Chapter 5 Meta-Analysis of European Heat Pump Field Trial Efficiencies

Introduction

This chapter attempts a meta-analysis of seasonal performance for ground-to-water and air to water heat pumps based on eight European trials for over 600 installations from five countries that provide thirteen different descriptors of performance. Seven of these trials have previously been published but no overview of their results has been attempted in terms of system boundary analysis. These reports provide information predominantly focused on means and ranges.

It has not been possible to assess and compare the uncertainty of the data from all eight trials since there was no common measurement methodology, no single set of instrumentation or shared assessment of precision and accuracy, moreover, the relevant information is not present in all reports. The data is presented here as it is presented in the various reports, no attempt has been made to verify or compare the level of uncertainty.

Trial boundaries represent one of the most important, if not the most important systematic source of apparent discrepancies between different empirical estimates of heat pump performance, and the only one that is capable of analysis outside the context of the field trial itself. The complexity of trial boundaries is not arbitrary, but reflects the real architecture of heat pump systems. This analysis is therefore capable of shedding significant light, not just on the measurement and analysis problem, but on the problems of designing effective heat pump systems.

Trial boundaries are rationalised from thirteen published descriptors to four values of seasonal performance providing the opportunity to reassess the UK EST heat pump trial results and identify two boundary conditions directly relevant to the interpretation of the Renewable Energy Sources Directive. What is apparent is the wide range in performance at all boundaries and in all trials indicating that heat pumps are sensitive to design and installation practice. The overarching theme is the need for a unified framework for reporting heat pump performance and its applicability to the re-analysis of existing data. The task of building such a framework has proved beyond the present author, but the work presented here represents an attempt to scope the potential value and the combination of analytical and practical difficulties that would need to be faced by those undertaking such a task.

This chapter is based on a published peer-reviewed paper (Gleeson, Lowe, 2013) originally presented for publication in January 2013. The writing of this chapter has provided the opportunity to tidy up, rework and extend the analysis and to present the results more fully especially in the light of the EU Commission decision of March 2013 (EC, 2013) to define the relevant boundaries applicable to the Renewable Energy Sources Directive.

Nomenclature

Measurement	Definition	Dimensions
SPF_H	Heat pump seasonal performance factor for heating	Dimensionless
SEPEMO	$SPF_{H1} = \frac{QH\_hp + QW\_hp}{EHW\_hp}$	Dimensionless
SEPEMO	$SPF_{H2} = \frac{QH\_hp + QW\_hp}{ES\_fan/pump + EHW\_hp}$	Dimensionless
SEPEMO	$SPF_{H3} = \frac{QH\_hp + QW\_hp + QHW\_bu}{ES\_fan/pump + EHW\_hp + EHW\_bu}$	Dimensionless
SEPEMO	$SPF_{H4} = \frac{QH\_hp + QW\_hp + QHW\_bu}{ES\_fan/pump + EHW\_hp + EHW\_bu + EB\_pump}$	Dimensionless
JAZ	$JAZ1 = \frac{QH\_hp + QW\_hp + Qbt\_pump}{ES\_fan/pump + EHW\_hp + Ebt\_pump}$	Dimensionless
JAZ	$JAZ2 = \frac{QH\_bt + QW\_hp}{ES\_fan/pump + EHW\_hp + Ebt\_pump}$	Dimensionless
JAZ	$JAZ3 = \frac{QH\_bt + QW\_hp + QHW\_bu + Qsh\_pump}{ES\_fan/pump + EHW\_hp + Ebt\_pump + Esh\_pump}$	Dimensionless
JAZ	$JAZ4 = \frac{QH\_bt + QW\_hp + QHW_{bu} + Qsh\_pump}{ES\_fan/pump + EHW\_hp + Ebt\_pump}$	Dimensionless
SPTRI	$SPFhps = \frac{QH\_hp + QW\_hp + QB\_pump}{ES\_fan/pump + EHW_{hp} + EB\_pump}$	Dimensionless
SPTRI	$SPFhs = \frac{QH\_hp + QW\_hp + QB\_pump + QHW\_bu}{ES\_fan/pump + EHW_{hp} + EB\_pump + EHW\_bu}$	Dimensionless
EST (SEFF)	$System\ Efficiency = \frac{QH\_hp + Qdhw\_tap + QB\_pump + QHW\_bu}{ES\_fan/pump + EHW_{hp} + EB\_pump + EHW\_bu}$	Dimensionless
EST SPF_{H5}	System Efficiency may be described as SPF_{H5}	Dimensionless
QH_hp	Quantity of heat of the heat pump in space heating (SH) operation	Wh or kWh
QW_hp	Quantity of heat of the heat pump in domestic hot water (DHW) operation	Wh or kWh
QHW_bu	Quantity of heat of the back-up heater for SH and DHW	Wh or kWh
QH_bt	Quantity of heat from the SH buffer	Wh or kWh
Qsh_pump	Quantity of useful heat from the SH pump downstream of the buffer tank	Wh or kWh
Qdhw_tap	Quantity of heat in DHW draw off (tapped hot water)	Wh or kWh

ES_fan/pump	Electrical energy use of the HP source: fan or brine/well pump	Wh or kWh
EB_pump	Electrical energy use of the heat sink (building) pumps for DHW and SH	Wh or kWh
EHW_hp	Electrical energy use of the heat pump for SH and DHW	Wh or kWh
EHW_bu	Energy use of back-up heater(s) for SH and DHW	Wh or kWh
Ebt_pump	Electrical energy use of the header circuit pump	Wh or kWh
Esh_pump	Electrical energy use of the SH pump downstream of the buffer tank	Wh or kWh

Table 5—1 Nomenclature adapted from SEPEMO

SPF and trial comparisons

EN 15316-4-2: 2008 defines SPF as:

> "the ratio of the total annual energy delivered to the distribution subsystem for space heating and/or domestic hot water to the total annual input of driving energy …. plus the total annual input of auxiliary energy."

It identifies the system boundary of "the heat pump generation subsystem" at these "distribution subsystems" as the heat flow to the space heating system and the heat flow in the tapped domestic hot water; its acceptance as a European Norm standard resulting in part from testing multifunctional heat pump systems by the International Energy Agency, see for example, Wemhoener, Afjei & Dott (2007).

However, field trial reports have been presented with no specific reference to EN 15316-4-2: 2008 and usually in isolation, without comparisons drawn between other trials and their boundary protocols. Studies that have focused on an overview of these various trial reports, a meta-analysis, have presented the results as either COP or SPF and, whilst recognising the impact of backup heaters and circulation pumps, have generally not compared them against a set of clear boundary definitions. Staffell (2009) provides a review of published trial efficiencies ranging from those for over 40 ground source installations through to individual units as well as heating season-only trials. These are presented with results quoted as either COP or SPF where COP values, "do not take into account any additional energy used on the backup heater" and where SPF is described as: "including any energy required and produced by the backup immersion heater". Colbourne's review (Colbourne, 2010) of the performance of electric heat pumps includes Staffell's work and adds a large trial by the Swiss Federal Office of Energy, as well as smaller studies from Austria, Sweden and France. Colbourne specifically refers to the impact of backup heaters in lowering seasonal efficiency whilst describing all results as SPF. Colbourne's work provides the air source heat pump efficiency data for Johnson's HFC impact study (Johnson, 2010) where SPF is defined as: "heat delivered/electrical energy input".

The first report to comment on the difference in boundary conditions between European trials is from Delta Energy and Environment in 2011. Focusing on the UK Energy Saving Trust trials between 2008 and 2010, it compares the results to those of the Fraunhofer Institute and the Swiss Federal Office of Energy. Delta comment:

> "Throughout the paper we refer to Seasonal Performance Factor (SPF) – effectively the average COP (co-efficient of performance) measured throughout the trial period. Note that due to differences in methodology between trials, the results are not completely comparable – due to the wider system boundary used in the EST trial, the UK results are likely to be lower (possibly by a SPF of around 0.1) than the other trials." (Delta, 2011)

Other published comments on the EST trials include: "relatively poor results from UK installations compared to European experience", and "heat pump performance in the UK is on average worse than in continental Europe" (Boait, Fan & Stafford, 2011). In the same vein: "evidence is emerging that heat pumps may be underperforming in the UK compared with other European countries" (Stafford & Lilley, 2012) and, "In particular, issues relating to the consistent definition of system boundaries are likely to prove difficult to resolve" (Stafford, 2011).

What is apparent from cross-European studies is the need to compare performance against a consistent boundary analysis. Provided that the implications of boundary identification are recognised, existing data should enable the assessment of the current mean for ground and air source heat pump installations and, importantly, the range of performance experienced in these trials. This requires an investigation into boundary conventions and assessment of appropriate boundary conditions to represent heat pump performance.

System Boundaries

European field trials can be broadly classified into two main boundary schemas based the German *Jahresarbeitzahlen* (JAZ) model, effectively in English, "seasonal performance factor" and the recent European model developed by SEPEMO-Build for seasonal performance factor classifications.

Jahresarbeitzahlen

Baumgartner, *et al,* (1993) provide a block diagram, later adapted by Wemhoner (Figure 5-1), representing three efficiency boundaries with the JAZ monovalent boundary *WPA-Wärmepumpenanlage* "heat pump system", comprising the source, the heat pump, circulation pump for the hydraulic header circuit and a buffer vessel, historically considered a necessary component for efficient heat pump operation. JAZ is measured at this boundary as the ratio of heat energy out (therefore minus any buffer vessel losses) to electrical energy in. Bivalent systems, again historically a boiler system, provide the next boundary. Finally, the inclusion of the space heating system and its circulation pumps provides a total system analysis. Wemhoner, et al, (2003) take this analysis and provide a JAZ boundary definition for IEA Annex 28 which includes all system circulation pumps, effectively including all energy inputs and outputs.

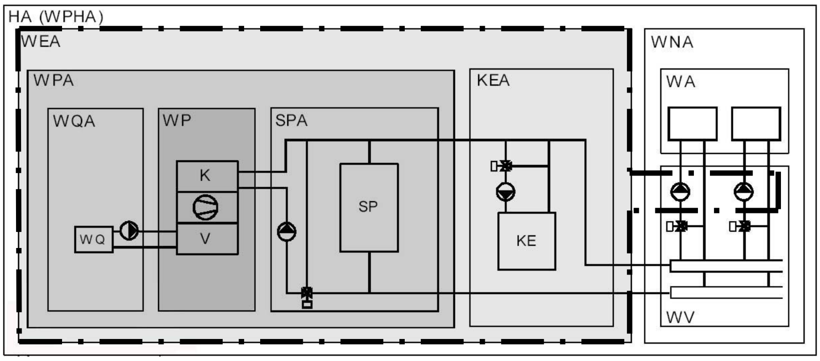


Figure 5—1 IEA Annex 28 JAZ definition (Wemhoner, et al, 2003 – after Baumgartner, et al, 1993)

The current German standard for assessing heat pump performance, VDI 4650: 2009, is the latest iteration in Jahresarbeitzahlen methodologies. A typical graphical depiction of Jahresarbeitzahlen is shown in Figure 5-2 where four separate JAZ boundaries have been defined that clearly relate to the Baumgartner/Wemhoner boundaries and where the higher index numbers indicate more components, greater complexity and progressively lower SPF efficiencies for the same heat pump installation.

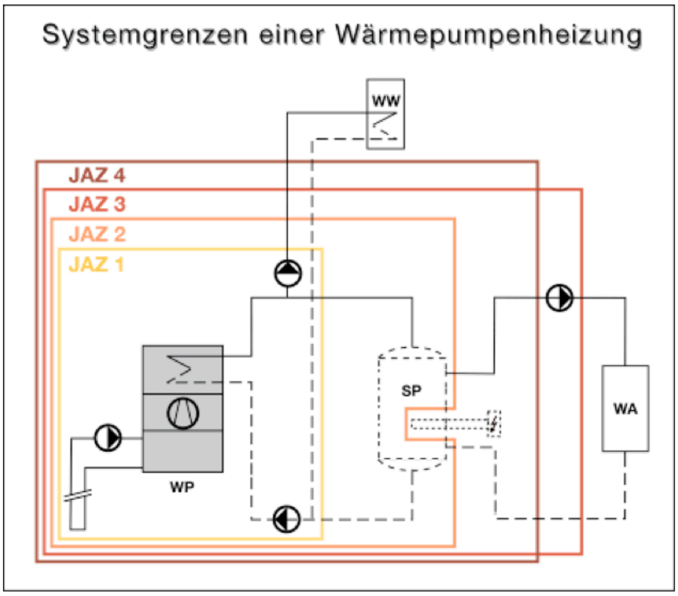


Figure 5—2 Jahresarbeitzahlen boundaries (online)

JAZ 1: *Erzeuger JAZ* or "after the heat pump" with source fans or pumps, controls and compressor plus what could be described as a header pumps – the circulation pump(s) to the distribution header feeding a buffer vessel and hot water cylinder.
JAZ 2: *System JAZ* or JAZ 1 plus space heating buffer storage losses, Baumgartner's JAZ.
JAZ 3: *Anlage JAZ* or "installation", comprising JAZ 2 plus any backup/boost heating and the space heating circulation pump; therefore Wemhoner's IEA Annex 28 definition.
JAZ 4: JAZ 2, minus the circulation pump. JAZ 4 may be applied where comparison is made with conventional boiler performance. Since the energy demand for space heating circulation pumps depends on system size factors such as volume flow and resistance, this is excluded in any comparison of heat source efficiency.

Where there is a single circulating pump or no buffer vessel, only the energy fraction used to circulate primary water to the domestic hot water cylinder should be included in JAZ 1 and JAZ 2 making monitoring programmes more complicated to set up. Similarly, it is apparent that header pump energy is dependent on circuit mass flow rate and resistance through the pipework and valves, condenser, buffer vessel and hot water storage cylinder, all functions of the individual installation and not of the heat pump itself. Jahresarbeitzahlen boundaries are reported for two field trials, the largest European trial from FAWA, Switzerland and a Local Agenda 21 programme from Lahr in Germany.

FAWA, Switzerland

The Swiss Federal Office of Energy report (Erb, et al, 2004) on their *Feldanalyse von Wärmepumpenanlagen* project "Field Analysis of Heat Pump Installations" or FAWA, is based on field trial data collated between 1995 and 2004 for 221 heat pumps at the JAZ 2 boundary. Some 50% of the installations included domestic hot water with 22% relying on the heat pump only. The trials cover both new build and existing housing with some 60% new and 40% of what is described as *Sanierungsobjekten* or "renovation projects". The range of building heat loss is from 28 to 208 kWh/m^2pa with a mean of 75 kWh/m^2pa due to the dominance of new build and low energy refurbishment. FAWA combine the data for both new and existing housing, with and without domestic hot water, to present a single JAZ 2 value, Table 5-2.

Ground Source Heat Pumps	Mean	Range	Air Source Heat Pumps	Mean	Range
JAZ 2	3.4	2.3 – 5.3	JAZ 2	2.6	1.5 – 4.0

Table 5—2 FAWA JAZ 2 mean and range

Lahr, Germany

The Lahr trial in the Black Forest region of Germany, undertaken through the Local Agenda 21 programme, collected data on 32 heat pumps including 12 air source, 7 water source and 13 ground source; the results published on a dedicated website, http://www.agenda-energie-lahr.de. The report write-up (Auer & Schote, 2009) provides results for "*Erzeuger JAZ*", JAZ 1, and "*System JAZ*", JAZ 2, Table 5-3.

LAHR		No	JAZ 1		JAZ 2		JAZ 2 All	JAZ 2 All
			Mean	Range	Mean	Range	Mean	Range
ASHP	Underfloor	7	2.8	2.3 – 3.2	2.4	1.9 – 2.8	2.3	1.7 – 3.0
	Radiators	5	2.4	1.9 – 2.8	2.2	1.7 – 2.6		
GSHP	Underfloor	11	3.4	2.0 – 4.4	3.1	2.3 – 4.2	3.1	2.3 – 4.2
	Radiators	2						
WSHP	Underfloor	6	3.2	2.0 – 4.2	2.9	-----	2.9	-----
	Radiators	1						

Table 5—3 Lahr Agenda 21 heat pump trial results

SPTRI, Sweden 2007

Early Swedish trials from 2007 by SP Technical Research Institute (Stenlund & Axell, 2007) provide a model with two boundaries, SPFhps and SPFhs. SPFhps refers to the seasonal efficiency of the heat pump system, the source pump and central heating sink pump at boundary A, Figure 5-3. SPFhs is the seasonal efficiency of the whole heating system including any backup heating at boundary E. The trial results are for 5 ground source heat pumps only, Table 5-4.

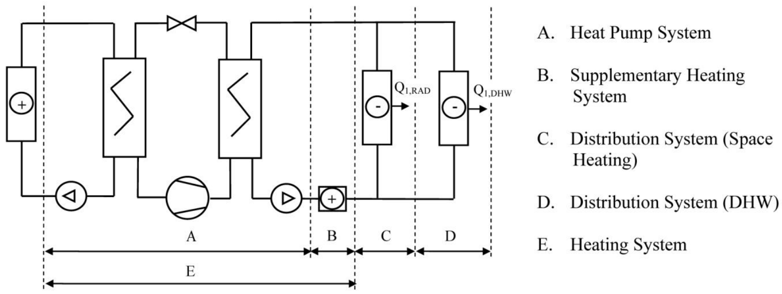


Figure 5—3 SPRTI GSHP boundary definition. (Stenlund & Axell. 2007)

GSHP	Mean	Range
SPFhps	2.9	2.5 - 3.1
SPFhs	2.6	2.4 - 2.9

Table 5—4 SPTRI GSHP trial results. (Stenlund & Axell. 2007)

The Association of Austrian Electricity Companies funded Arsenal Research to develop, test and evaluate a "standardised monitoring methodology" for heat pump systems (Huber & Glasner, 2007). The resulting SPF is the ratio of heat energy out (heat sink) to energy in (compressor, source and sink circulation pumps), the same SPFhps boundary defined by SPTRI, Figure 5-4. Unfortunately published results are for direct expansion ground loops only and thus, for technological rather than methodological reasons, not in principal comparable with brine filled ground source heat exchangers or air source heat pumps.

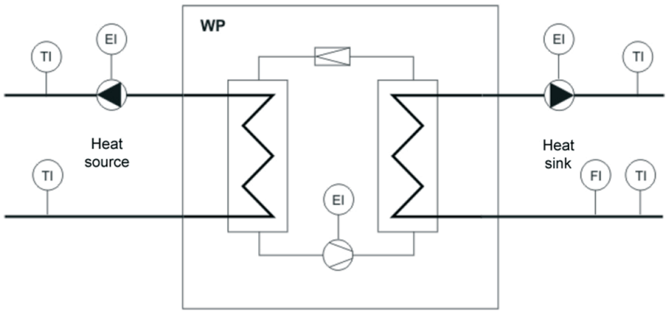


Figure 5—4 Arsenal Research standardised monitoring methodology (Huber & Glasner, 2007)

UK EST Trials, 2009-2010

The UK Energy Savings Trust (EST, 2010) has uniquely applied the concept of "System Efficiency", with the inclusion of hot water draw off within the overall SPF heating system boundary. The UK Department of Energy and Climate Change (DECC) have published an updated report on the EST trial (Dunbabbin & Wickins, 2012), which includes a both a boundary image, Figure 5-5, and updated results for 71 installations, Table 5-5.

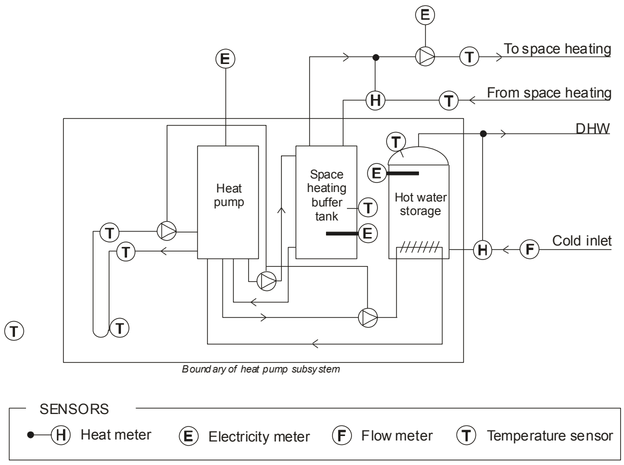


Figure 5—5 System Efficiency (Dunbabbin & Wickins, 2012)

System efficiency	Ground source	Air source
Number	49	22
Average	2.39	1.82
Range	1.55 – 3.37	1.2 – 2.2

Table 5—5. EST Field Trial System Efficiencies (Dunbabbin & Wickins, 2012)

Whilst the Figure 5-5 boundary from DECC excludes the space heating circulation pump, this pump was included as an energy input in published results. There are advantages and disadvantages to measuring hot water draw off when comparing the efficiency of regular boilers to combination boilers or instantaneous water heaters. However, the heat pump is heating a hot water cylinder and cylinder heat loss is dependent primarily on volume of storage, insulation, temperature difference between stored water and ambient and the rate of hot water use, none of which are related to the efficiency of the heat pump. Where hot water is heated but not used, cylinder losses are included in the denominator whether useful or not. The EST condensing boiler trials (Orr, et al, 2009) included draw off within the system boundary and, it is assumed that in accordance with this historical precedent, the EST heat pump trials also included draw off. The EST approach to heat pump monitoring, using "System efficiency", has raised some concerns regarding comparability with other heat pump trials although there was at the time no internationally agreed trial methodology. However, the "System efficiency" approach is in the spirit of EN 15316-4-2: 2008 and therefore has much to recommend it.

SEPEMO boundaries

The profusion of monitoring methodologies and the confusion over appropriate boundary setting is the subject of the European Union Intelligent Energy Europe research project: "SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector" (SEPEMO-Build). The earliest published report specifically aimed at considering field trial system boundaries from a common European approach was published in 2010 by SP Technical Research Institute, Sweden (SPTRI) (Nordman, et al, 2010), the lead partner for SEPEMO. The SEPEMO project published a detailed analysis in 2011 (Zottl & Nordman, 2011) providing four boundary definitions and their equations. The SEPEMO methodology consists of the heat pump only (SPF_{H1}), with expanding boundaries covering fan or pump power into the heat pump (SPF_{H2}), back up heaters (SPF_{H3}) and finally, system circulators or pumps (SPF_{H4}). Note again the relationship between the higher index number and lower SPF efficiency for the same installation. Extending the SEPEMO boundary approach, the inclusion by the EST of tapped hot water, "System efficiency", rather than heat into the hot water cylinder, could be defined as SPF_{H5}, shown as the outer dotted boundary, Figure 5-6.

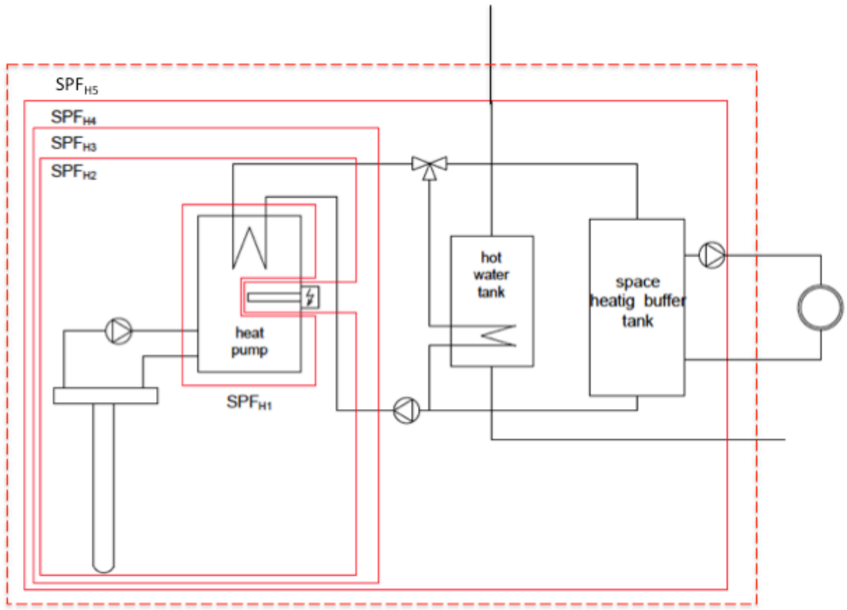


Figure 5—6 SEPEMO system boundaries (after Zottl & Nordman, 2011)

SPF_{H1} is impractical as a field measurement since there is no means to drive heat into or out of the heat pump, therefore, practical measurements must take place at any of the boundaries defined as SPF_{H2}, SPF_{H3} or SPF_{H4}. It is important to note that where backup is by immersion heater to the hot water cylinder the energy supplied is included in both numerator and denominator for SPF_{H3} and SPF_{H4}.

SEPEMO boundaries have been applied to field trials by the Fraunhofer Institute in Germany, the Danish Technology Institute (DTI) and SPTRI Sweden. The boundary definitions for the Fraunhofer and DTI trials differ from those of SEPEMO only by boundary numbering. Where SEPEMO define the heat pump alone as SPF_{H1}, Fraunhofer and DTI describe this as SPF 0, the expanding boundaries therefore differ by 1. Fraunhofer results are generally quoted for SPF 2 (SPF_{H3}), whereas DTI publish only SPF 3 (SPF_{H4}).

SPTRI Sweden 2010

The SPTRI (Nordman et al, 2010) report on seven ground source heat pumps is important due to its direct link to the SEPEMO director, Nordman, and its use of the SEPEMO boundary methodology; results are given for SPF_{H1} and SPF_{H3}, Table 5-6.

SPTRI 2010	Heating & DHW	Heating & DHW	Heating only	Heating only
GSHP	SPF_{H1}	SPF_{H3}	SPF_{H1}	SPF_{H3}
Mean	3.7	3.26	4.6	4.2
Range	2.7 - 4.1	2.6 - 3.6	3.9 - 5.4	3.4 - 5.1

Table 5—6 SPI Sweden GSHP SPF

SPF_{H1} shows the impact of sink temperature is reflected in the drop in efficiency of 18% between heating only and heating and DHW. An SPF_{H3} drop in efficiency of 23% is associated with the inclusion of DHW.

Fraunhofer existing buildings trial

The Fraunhofer Institute (Russ, et al, 2010) report on 72 heat pump systems in existing buildings, *Gebäudebestand*, all with domestic hot water, Table 5-7. The trial investigates 36 ground, 34 air and 2 water source heat pumps. These are further subdivided into 10 ground collector (ground loop) and 26 borehole, 21 air source and 13 exhaust air source.

Ground Source Heat Pumps	Mean	Range	Air Source Heat Pumps	Mean	Range
SPF_{H3}	3.3	2.2-4.8	SPF_{H3}	2.6	2.1-3.4

Table 5—7 Fraunhofer Existing Buildings mean SPF

The house sizes range from 90 to 360m^2 with a mean of 181m^2. The range of heat losses are from 85 to 340 kWh/m^2pa.

Fraunhofer new buildings trial

The Fraunhofer Institute (Miara, et al, 2011[a & b]) report on field trials in new buildings for space heating and domestic hot water, carried out between 2005 – 2010, on 110 heat pumps with a final report on 56 ground-to-water, 18 air to water and 3 water-to-water. Underfloor heating was installed in all installations but one, and weather compensation was used throughout. Applying SEPEMO definitions, results are published for mean performance at SPF_{H1}, SPF_{H2}, SPF_{H3} and SPF_{H4}, with Delta referring to SPF 2 means, that is, SEPEMO SPF_{H3}. Fraunhofer 'new build' results at SPF_{H3} are 3.9 for ground source and 2.9 for air source, Table 5-8.

Ground Source Heat Pumps	Mean	Range	Air Source Heat Pumps	Mean	Range
SPF_{H1}	4.19		SPF_{H1}	3.17	
SPF_{H2}	3.93		SPF_{H2}	2.95	
SPF_{H3}	3.88	3.1 – 5.1	SPF_{H3}	2.89	2.3 – 3.4
SPF_{H4}	3.75		SPF_{H4}	2.74	

Table 5—8 Fraunhofer New Build mean SPF

Comparing Fraunhofer ground and air source from the "existing" buildings to those of the "new" provides means in existing buildings of 3.3 and 2.6, compared to the values for new build at 3.9 and 2.9, a percentage reduction of 10 and 13% respectively; a result partially explained by the use of radiators for 71% of the existing building heating systems.

Danish Technological Institute

The Danish Technological Institute (DTI) (Pederson & Jacobsen, 2011) report on 170 field trials monitored between the period May 2010 and July 2011 on both new and existing dwellings. The final results included 138 ground and 12 air source, some 20 heat pumps being excluded from the final report due to, "data reliability and their analysis", including the exclusion of heat pumps whose "COP" exceed 5.5 (4 units) or fell below 1.5 (8 units). The trial results are based on the Fraunhofer SPF designations and given for SPF 3 (SPF_{H4}), that is, a boundary including backup heater and circulation pump. As would be expected from the sample size, the data for the ground source heat pumps is instructive, with a 17% reduction in SPF between radiators and underfloor. Whilst less instructive, the air source data provides a similar pattern relating SPF to emitter type, Table 5-9. Importantly, the DTI data provides a difference in SPF between air and ground source heat pumps operating on mixed emitters of 23%. Weighted averages for all emitters provide means of 3.03 and 2.33 with ranges of 3.1 to 5.1 and 2.3 to 3.4 for ground and air source respectively, Table 5-9.

GSHP	Mean	Range	ASHP	Mean	Range
SPF_{H4} Radiators	2.72		SPF_{H4} Radiators	2.14	
SPF_{H4} Underfloor	3.04		SPF_{H4} Underfloor	2.34	
SPF_{H4} Mixed	3.27		SPF_{H4} Mixed	2.87	
SPF_{H4} All	3.03	3.1-5.1	SPF_{H4} All	2.33	2.3-3.4

Table 5—9 DTI SPF_{H4} for radiators, mixed radiators & underfloor, underfloor

Analysis of reported boundary conditions

Thirteen boundary descriptors are met in eight heat pump trials lasting for a minimum of one year and consisting of over 600 heat pump installations. The review of these methodologies indicates that seven of these boundaries are unique: JAZ 1, JAZ 2, SPFhps, SPF_{H1}, SPF_{H2}, SPF_{H3}, SPF_{H4} and SPF_{H5} with six descriptors being redundant. The highest practical SPF is achieved by SPF_{H2} followed by JAZ 1, the inclusion of the header circulation pump. JAZ 2 takes into account both the header pump and any losses from any buffer vessel should one be installed. SPF_{H3} includes only the backup heater and not the sink pump. SPFhps includes the total energy demand of the sink circulation pump, whereas JAZ 3, SPFhs and SPF_{H4} are identical and include both integrated backup heating and all circulation pumps. Finally, “system efficiency” provides the resulting SPF_{H5} for all loads including domestic hot water cylinder losses.

Arguments may be made for the exclusion of backup heaters where installation issues such as undersizing of the heat pump or poor heat distribution, neither of which are directly associated with heat pump efficiency, will require additional heating. However, it should be considered legitimate to include the backup required for heat pumps incapable of reaching domestic hot water pasteurisation temperatures, a situation exemplified by most domestic HFC-based units. The inclusion of hot water cylinder losses is an extension of the logic that includes buffer vessel losses and is entirely dependent on size, insulation and demand. When considering the same heat load, comparing a heat pump driven wet central heating system to, for example, electric storage heaters or warm air heating, JAZ 3 or SPF_{H4} is the logical choice since these boundaries capture all system energy inputs including the circulation pump to the space heating.

The bulk of the data is from five trials: FAWA, EST, Fraunhofer ‘new’ and ‘existing’ and the DTI. Since the boundaries used by the Fraunhofer Institute, the Danish Technical Institute and SPTRI 2010 are identical they may all be renamed using SEPEMO terminology. SPFhs is identical to SPF_{H4}, that is, all system electrical inputs. The boundaries applied to the different trials are shown in Table 5-10, the total number of installations in Table 5-11 where all SEPEMO-based boundaries comply with SEPEMO nomenclature.

Boundary	FAWA	Lahr	SPTRI 2007	SPTRI 2010	EST	Fraunhofer Existing	Fraunhofer New	DTI
JAZ 1		✓						
JAZ 2	✓	✓						
SPFhps			✓					
SPF 0/$_{H1}$				✓			✓	
SPF 1/$_{H2}$							✓	
SPF 2/$_{H3}$				✓		✓	✓	
SPFhs/SPF 3/$_{H4}$			✓				✓	✓
System Efficiency/SPF$_{H5}$					✓			

Table 5—10 Trial boundary analysis

Rationalising all SPF boundaries to comply with SEPEMO nomenclature provides Table 5-11.

TRIAL	N^o	JAZ1	JAZ2	SPFhps	SPF$_{H1}$	SPF$_{H2}$	SPF$_{H3}$	SPF$_{H4}$	DHW
FAWA	221		221						50%
Fraunhofer new	74				74	74	74	74	100%
Fraunhofer existing	70						70		100%
DTI	150							150	100%
LAHR	25	25	25						unknown
SPTRI 2007	5			5				5	100%
SPTRI 2010	6				6		6		86%
EST	71								77%
Total	622	25	246	5	80	74	150	229	

Table 5—11 Total numbers of heat pump at the various boundaries

Further boundary compression

For a meta-analysis of heat pump performance we may wish to re-analyse the significant work of FAWA, Lahr or the EST in SEPEMO terms, or, for example, to recalculate DTI results from SPF$_{H4}$ to SPF$_{H3}$, in order to compare all data in the same boundary category. Even if we assume 100% heat transfer into the system from header pumps, buffer vessels, backup and circulation pumps, there is no mathematical approach that will separately identify these impacts from the reported arithmetic mean efficiencies since each additional input introduces an unknown quantity of heat into the efficiency equations. In reality these electrical inputs will not provide 100% useful heat transfer and the resulting equations introduce yet more unknowns. Some grasp of the challenge may be apparent from the equations representing Fraunhofer New and Lahr GSHP trials, Equations 5-1 and 5-2.

$$SPF_{H2} = \frac{QH\_hp + QW\_hp}{ES\_fan/pump + EHW\_hp} = 3.93$$

$$SPF_{H3} = \frac{QH\_hp + QW\_hp + QHW\_bu}{ES\_fan/pump + EHW\_hp + EHW\_bu} = 3.88$$

$$SPF_{H4} = \frac{QH\_hp + QW\_hp + QHW\_bu}{ES\_fan/pump + EHW\_hp + EHW\_bu + EB\_pump} = 3.75$$

Equation 5-1. Comparison of SPF means

$$JAZ1 = \frac{QH\_hp + QW\_hp + Qbt\_pump}{ES\_fan/pump + EHW\_hp + Ebt\_pump} = 3.4$$

$$JAZ2 = \frac{QH\_bt + QW\_hp}{ES\_fan/pump + EHW\_hp + Ebt\_pump} = 3.1$$

Equation 5-2. Comparison of JAZ means

SPF_{H2} differs from JAZ1 solely due to the header circuit pump. This pump consumes an unknown quantity of electricity, Ebt_pump, and transfers an unknown fraction of the electricity as heat to the installation water, Qbt_pump. Similarly, QH_bt, the useful output from the buffer vessel is an unknown fraction of QH_hp, the heat energy entering the buffer vessel. Without access to the trial raw data, it is not mathematically possible to uniquely transpose trial arithmetic means in order to recalibrate between trial results either in JAZ or SPF units; of necessity, a qualitative approach would be required.

EST re-analysis

The EST trial results have caused some consternation in the UK where this field trial has provided evidence, for some observers, of poor 'as-installed' heat pump performance. However, the trial was carried out to assess the state of heat pump installation in the UK and, positively, has proven to be the catalyst for critical reflection on installation practice and the production of extensive guidance through the Microgeneration Certification Scheme heat pump design guide MIS 3005 (DECC, 2012) and other supporting documents.

Due to data logger/meter location, the some of the trial raw data can provide SEPEMO or SPFhps related outputs. Differentiating between boundaries requires, in some instances, allowances for integrated central heating pumps and hot water cylinder heat losses. Most of the heat pumps used in the EST trials had integrated circulation pumps and would therefore most closely emulate either the SPFhps, no back up, and SPF_{H4}, all inputs. The following analysis is based on raw data only and, since no heat balance has been carried out, the results are therefore provisional and unconfirmed.

Many of the EST installations are space heating only and readily fit into boundary categories other than System Efficiency, SPF_{H5}. A review of the EST raw data for 52 ground source installations provides SPF values for 9 heat pumps at SPF_{H2}, 10 at SPFhps, 17 at SPF_{H4} and 41 at SPF_{H5} which can be compared to the DECC results where the efficiency of all installations is described as System Efficiency, Table 5-12.

GSHP	No	Mean	Range
SPF_{H2}	9	2.6	1.9 – 3.3
SPFhps	10	2.4	1.9 - 3.2
SPF_{H4}	17	2.5	1.4 - 3.3
SPF_{H5}	41	2.3	1.5 – 3.4
DECC	49	2.4	1.6 – 3.4

Table 5—12 EST GSHP Field Trial

The same analysis of 24 air source installations provides SPF values for 4 heat pumps at SPF_{H2}, 9 at SPFhps, 7 at SPF_{H4} and 12 at SPF_{H5}, Table 5-13.

ASHP	No	Mean	Range
SPF_{H2}	4	2.9	2.2 – 4.0
SPFhps	9	2.3	1.9 – 2.6
SPF_{H4}	7	1.9	1.2 – 2.3
SPF_{H5}	12	1.9	1.5 – 3.0
DECC	22	1.8	1.2 – 2.2

Table 5—13 EST ASHP Field Trial

This analysis casts a somewhat different light on the EST trial results indicating that trial mean SPF is dependent on system boundary definition. If the trial results constitute a sample, we now have sub-samples corresponding to each system boundary that is present in the sample. For very small sub-samples, such as SPF_{H2}, the calculated mean is vulnerable to the impact of possible outliers. The Energy Savings Trust's use of System Efficiency is unique and its reclassification, where possible, as SPFhps and SPF_{H4} is certainly more useful when comparing trial outputs.

Combining all trial results

As noted earlier, SPF_{H1} excludes the source fan/pump, sink pump and any backup and therefore, since it is not indicative of 'real world' operation, may be removed. Lahr alone provides JAZ 1, with values for 23 heat pumps, 13 ground and 12 air source. EST alone provide "system efficiency" and since the objective is to compare trial results these may be omitted.

JAZ 2 and SPFhps are similar in that that JAZ 2 includes the header circulation pump and buffer losses whereas SPFhps includes the full sink pumping requirements only; SPFhps may be reclassified as JAZ 2 without any great loss of accuracy. JAZ 2 from FAWA, Lahr, SPTRI 2007 and EST provides 274 heat pumps, the largest classification.

Removing JAZ 1, SPF_{H1}, SPFhps and SPF_{H5} reduces boundary classifications from eight to four: JAZ 2, SPF_{H2}, SPF_{H3} and SPF_{H4}. The boundaries may now be analysed by source to provide number, weighted mean and range, Tables 5-14 and 5-15.

GSHP	JAZ 2			SPF_{H2}			SPF_{H3}			SPF_{H4}		
TRIAL	Nº	Mean	Range	Nº	Mean	Range	Nº	Mean	Range	Nº	Mean	Range
FAWA	100	3.4	2.3 - 5.3									
LAHR	25	3.1	2.3 - 4.2									
SPTRI 2007	5	2.9	2.4 - 2.9							5	2.6	2.4 - 2.9
EST	10	2.4	2.4 - 3.5							17	2.5	1.4 - 3.3
SPTRI 2010							6	3.26	2.6 - 3.6			
Fraunhofer Existing							36	3.3	2.2 - 4.8			
Fraunhofer New				56	3.93		56	3.88	3.1 - 5.1	56	3.75	
DTI										138	3.03	3.1 - 5.1
No, mean, range	140	3.3	2.3 - 5.3	56	3.9		98	3.6	2.2 - 5.1	216	3.2	1.4 - 5.1

Table 5—14 GSHP Meta-analysis

ASHP	JAZ 2			SPF_{H2}			SPF_{H3}			SPF_{H4}		
TRIAL	Nº	Mean	Range	Nº	Mean	Range	Nº	Mean	Range	Nº	Mean	Range
FAWA	100	2.6	1.5 – 4.0									
LAHR	25	2.3	1.7 - 3.0									
EST	9	2.3	1.9 - 2.6							7	1.9	1.2 - 2.3
Fraunhofer Existing							34	2.6	2.1 – 3.4			
Fraunhofer New				18	2.95		18	2.89	2.3 – 3.4	18	2.74	
DTI										12	2.33	2.3 – 3.4
No, mean, range	134	2.5	1.5 - 4.2	18	3		52	2.7	2.1 – 3.4	37	2.4	1.2 – 3.4

Table 5—15 ASHP Meta-analysis

Discussion

Changes to heat pump manufacture and quality of installation are crucial in understanding, in particular, the range of performances. Because of its historical overview, the FAWA report provides evidence of the increasing efficiency of heat pumps (Eschmann, 2004):

> "Between 1995 and 2003, the SPF improved by approximately 20% for both groups. Since the start of the project, the SPF data for 59% A/W [air to water, i.e., ASHPs] and 41% B/W heat pumps [brine to water, i.e., GSHPs], plotted against the total installed Swiss heat pump capacity, show an increase of around 23% (from 2.5 to 3.1)."

This increased efficiency may be partly ascribed to the increased efficiency of manufacturers' heat pump units, evident from the laboratory COP test data from WPZ, Buchs, Switzerland, Figure 5-7, allied to improvements in system design as Swiss knowledge of the technology has matured. The general improvement in heat pump design would imply that recent installations should be more efficient.

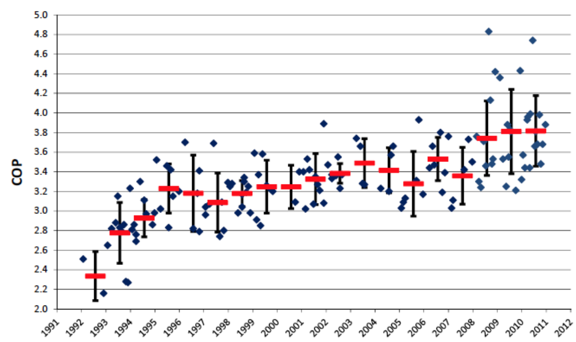


Figure 5—7 COP historical improvement. (WPZ)

Tables 5-14 and 5-15 show that Fraunhofer new building installations have the highest efficiencies. This is due to designing, wherever possible, for minimal use of backup heaters, for low temperature space heating with weather compensated control and variable speed circulation pumping, all of which produce the lowest energy input to output ratio. Combining both Fraunhofer New and Existing results allows for comparison with a contemporary study of both new and existing buildings in a mature market, that of Denmark. The combined Fraunhofer trials produce weighted SPF_{H3} mean values of 3.7 and 2.7 for ground and air source respectively. Although not directly comparable to the DTI SPF_{H4} values of 3.03 and 2.33, they do indicate that even when adjusted for circulation pump, about a 10% reduction[9], the Fraunhofer installations still outperform the Danish.

In addition to their comments on SPF boundary, of "System Efficiency", Delta raise the following points concerning the EST field trials:

> "Other important differences to note, which may contribute to lower SPFs in the UK, are:
>
> The German and Swiss heating systems are typically of higher quality than those in the UK (in terms of the quality of components and control system).
>
> UK and German installations were providing a higher proportion of DHW than in Switzerland.
>
> UK buildings were (broadly) of lower quality in terms of insulation / rate of heat loss.
>
> These issues may have reduced achievable SPF in the UK by a few percentage points, but these factors alone are not sufficient to explain the UK trial's poorer results."

We may offer the following comments supported by Tables 5-14, 5-15 and 5-16:

- Heat pumps are subject to the same market conditions as any other commodity leading to the use of similar manufacturing techniques, materials, components and international marketing strategies. Some of the manufacturers in the UK trials also appear in the German and Danish trials. A significant difference is in system design where weather compensation control in particular and variable speed pumps have a far higher market penetration in continental Europe than in the UK.
- Some 50% of Swiss installations (FAWA) have a DHW load, EST have around 70% whereas all Fraunhofer installations produce DHW. If DHW has a significant impact on SPF then the UK results should lie between the two. Apart from ASHPs at JAZ 2, all UK heat pumps underperform by at least 20% with respect to the boundary group means.
- The DTI state that their installations encompass both new and existing buildings although there is no information on the actual heat losses. FAWA describe both new and refurbished with a mean of 75 kWh/m^{2}pa and a range between 28 and 208 kWh/m^{2}pa. Fraunhofer Existing provide a mean of 177 kWh/m^{2}pa with a

[9] The impact of the circulator pump is dependent on the mass flow rate of water, the hydraulic system pressure losses due to friction and the pump design and its operation. Low energy pumps have power requirements of 40% or less than that of older models and variable speed control will further reduce pumping power. A 10% reduction in SPF due to the inclusion of the pump is thus an approximation.

range between 85 and 340 kWh/m^{2}pa. There is no heat loss overview in EST trial publications, however, some 20% of the dwellings are built post-2000. To put this in context, the average space heating load for UK dwellings is about 180 kWh/m^{2}pa (Dowson, 2012) and thus the Fraunhofer trials of existing buildings, with higher SPFs than those of the EST, cannot reasonably be said to focus on low energy refurbishments.

- Mean SPF is a ratio of output to input and whilst high heat losses will result in higher fuel input (as with any space heating system), SPF is fundamentally dominated by Carnot source and sink absolute temperatures. There is a practical limit to output from underfloor heating due to floor surface temperature leading to high loss buildings requiring radiators. 20% of UK buildings used underfloor heating alone, 16% of Danish and only 3% of Fraunhofer existing.
-

TRIAL	UFH	Mixed	Radiator	DHW	New build	Mean Heat loss kWh/m2pa	Range kWh/m2pa
DTI	16%	70%	14%	100%	N/K	N/K	N/K
EST	21%	14%	64%	73%	19%	≈ 90	N/K
Fraunhofer existing	3%	26%	71%	100%	0%	177	85 - 340
FAWA	93% total, 53% refurb*	N/K*	N/K*	50%	60%	75	28 - 208

* FAWA state: "92% of cases of buildings have underfloor heating that is partially complemented by radiators. In renovation projects, the proportion of underfloor heating systems is 53%." K/N not known.

Table 5—16 Comparison of the different trial installation data

High envelope heat losses cannot explain the low SPF results from the UK when compared with comparable losses in the Fraunhofer existing buildings trial. High heat losses impact on system design, requiring higher temperatures from emitters, yet the EST trials have more underfloor systems than both DTI and Fraunhofer existing. If the issue is not the heat pump model, envelope losses or emitters then perhaps it is the quality of installation, a general underachievement in praxis. The wide range of performance identified in Tables 5-14 and 5-15 indicates the need for in-depth individual system analysis. The interpretation of measurements for heat pumps will depend on factors other than system boundaries, for example, monitoring intervals, completeness of datasets and treatment of errors. Unfortunately such detailed information on monitoring specifications is not available for several of the field trials referred to in this paper. It has therefore not been possible to include consideration of these questions in the comparison of heat pump performance across field trials.

However, the inconsistency in performance does raise the issue of design and installation competency and therefore an opportunity to readdress vocational education and training (VET) including design, matching heat pump to load, system installation, installation controls, the quality of system monitoring and the monitoring and analysis protocols.

Summary

This chapter is an exploration of the importance of system boundaries in the measurement and reporting of heat pump performance data. It begins by describing the system boundaries that have been used by major heat pump trials over the last 20 years. It goes on to demonstrate the impact of choice of boundary on the values of SFP that are quoted in the different studies.

The chapter then shows that a combination of analytical and practical redundancy allows considerable reduction (by roughly half) in the set of boundary conditions that need to be considered in analysing data but that, in the absence of trial raw data, significant, irreducible, differences remain between the four historical and contemporary definitions of JAZ 2, SPF_{H2}, SPF_{H3} and SPF_{H4}.

The chapter presents a short exploration of the possibility of introducing corrections to allow data for these remaining system boundaries to be mapped onto each other. The conclusion of this exercise is that uncertainties around the physical properties of sub-systems (heat stores, circulation pumps and electrical resistance heaters) which are either unmeasured or unreported in most of the studies examined, mean that such corrections are unreliable and, in the view of this author, of little value.

The final section is an attempt to reconcile one recent study of heat pumps, the EST field trial undertaken in the UK between 2009 and 2010, with the body of work undertaken in continental Europe. This exercise shows how careful analysis of boundary conditions can impact significantly on the conclusions from such comparisons.

The meta-analysis indicates that SPF_{H2} is the most relevant metric for heat pump efficiency when evaluating the RES 2009, $SPF > 1.15 \times 1/\eta$, since it includes only the source fan/pump, compressor and control electrical energy inputs and is directly comparable to alternative heat sources such as condensing gas boilers. However, SPF_{H2}

applies only to monovalent designs where the heat pump is sized to provide all necessary heat demand rather than rely on resistance backup for either space heating or domestic hot water. Where bivalent heat pump systems are installed, SPF_{H3} is the relevant metric for direct comparison to other forms of wet central heating.

The Commission Decision of 1 March 2013 (EC, 2013) has established the as-installed measured performance boundary to be SPF_{H2}, that is, the heat pump without backup or distribution circulating pumps. The Decision provides minimum performances based on three, somewhat poorly defined European climate zones, Figure 5-8, and default values for H_{HP} (annual equivalent heat pump hours) and SCOPnet (SCOP without back up) or SPF_{H2}, Table 5-17. It is immediately apparent that SCOPnet, as a laboratory test, is not subject to the vagaries of design and installation as is SPF_{H2}. In addition, since SCOPnet is derived from EN 14825:2012 part load COP testing, the pump/fan power is based only on pressure drop in the evaporator and condenser. The collection of renewable data based on SCOPnet would alleviate installers from the need for monitoring systems and save the costs associated with the data logging, data collection and analysis. RES outputs could be supplied by reference to heat pump sales rather than as-installed SPF_{H2}.

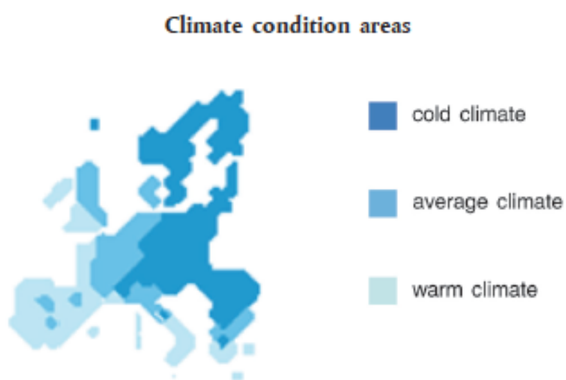


Figure 5—8 European climate zones (EC, 2013)

	Warmer climate		Average climate		Colder climate	
Heat pump type	H_{HP}	SPF_{H2}	H_{HP}	SPF_{H2}	H_{HP}	SPF_{H2}
Air to water	1170	2.7	1640	2.6	1710	2.5
Ground-to-water	1340	3.5	2070	3.5	2470	3.5
Water-to-water	1340	3.5	2070	3.5	2470	3.5

Table 5—17 Default values for electrically driven heat pumps (EC, 2013)

For the UK, the Commission decision has been interpreted by DECC in their EST Phase 2 report as:

> "The European Commission states that the minimum level of SCOP for a heat pump to be considered renewable is 2.5. The same document indicates that the system boundaries for this calculation are those of SPF_{H2}."

We note from the map that the correct values should be 2.6 and 2.7 for air and 3.5 for ground source heat pumps.

Given that manufacturers provide similar heat pump technologies the extreme ranges of performance, even in recent trials, confirms the sensitivity of heat pump performance. The current mean for air source heat pumps provides reasonable evidence that this technology will not meet the expectations of RES 2009 to provide renewable heat unless there is a significant change in electricity grid fuel mix, an increase in renewables to raise the value of η (eta), or significant inroads are made across Europe to raise the standards of design, installation and operation in order that the mean reflect the higher range values identified.

This sensitivity is the subject of the next chapter where EST field trial data provides the opportunity for a more forensic analysis than that possible from published trial mean.

Chapter 6 EST Trials 2009 -10: Evidence from operation

Introduction

The meta-analysis provided a comparison of different European trial means and ranges and, whilst useful, is not focused on the physical differences between individual heat pump systems that remain hidden where there is no access to trial raw data. Whichever way we define efficiency, the outcomes of both the German Lahr and UK EST trials have been seen as disappointing when compared to other trials. This chapter attempts to understand the EST trial raw data through simplified methods such as 'exploratory data analysis' (Tukey 1977, du Toit et al 1986, Myatt 2007) in order to identify macro issues and to subject them to a deeper analysis using a combination of quantitative and, where appropriate, qualitative data analysis tools. As noted in the methodology chapter, the technical monitoring specification did not call for a full energy balance to be carried out to validate the data received from the field trial. In addition, the position and accuracy of meters and sensors is unknown since the final decision on such equipment lay with the contractors and whilst there are methods for approximating missing data, for assessing data quality and for data cleansing, it was felt by the author that the absolute exactness of the data was secondary to the wider design issues raised by a taxonomic analysis; a full statistical treatment is still awaited.

Preface on system boundaries

Individual EST heat pump systems have been analysed in depth in the Phase 1 trial report (DECC, 2012) and the report peer reviewed (although unacknowledged) before publication by this author. The DECC 2012 report focuses on the physical operation of a large sample of the heat pumps including compressor run times and temperatures, the source and sink temperatures and ground and system circulator pump energy where available. DECC present efficiency under the classification of seasonal efficiency or SEFF however, due to the variation in metering positions, a range of boundary efficiencies may be determined from the trial raw data. These have been collated for both ground and air source in order to prepare the ground for a broader analysis, Tables 6-1 and 6-2.

GSHP

CODE		SPFH2	SPFhps	SPFH4	SPFH5	SEEF
407	oversized rads				2.1	2.1
408	oversized rads				2.2	2.2
409	oversized rads				2.0	2.0
410	oversized rads				2.0	2.0
411	oversized rads				2.3	2.3
412	oversized rads				1.9	1.9
413					2.3	2.3
414	WSHP Pool				2.7	2.7
415					2.4	2.4
416					2.3	2.3
417					2.6	2.6
419					3.0	3.0
420					3.0	3.0
421					1.7	1.7
430			2.3	2.3		2.3
431				2.8		2.8
432					1.8	1.8
433					1.8	1.8
434					2.0	2.0
435					2.2	2.2
436					1.8	1.8
437					2.3	2.3
438					2.1	2.1
439					3.4	3.4
451				2.3	2.2	2.2
452				3.3	3.0	3.0
453					3.3	3.3
454	oil no input only output				2.5	2.5
455					2.4	2.4
456	v18 shows input				2.2	2.2
457		2.5	2.5	2.5		2.5
458	Secondary circ				2.5	2.5
459					2.9	2.9
460		2.7	2.5	2.5	2.2	2.2
461		2.7	2.5	2.5	2.1	2.1
462		2.1	2.0	2.0	2.1	2.1
463		3.2	3.0	3.0	3.0	3.0
464		2.7	2.5	2.5	2.3	2.3
465		2.0	1.9	1.9	2.0	2.0
466					2.7	2.7
469	Data complete	2.1	1.9	1.7	1.6	1.6
470		3.3	3.2	3.2		3.2
471	4 months missing data			2.3	2.1	2.1
476					1.5	1.5
480					3.0	3.0
481	Boost to buffer				2.4	2.4
482	Boost to buffer				1.8	1.8
491					2.5	2.5
492	Independent DHW		3.4	3.4		3.4
927	427+428 24% of work in (v16) missing	5.6	5.0	4.8	4.7	4.7
950	450+468 20% of 450 SH (v21) missing				1.5	1.5
	MAX	5.6	5.0	4.8	4.7	4.7
	MIN	2.0	1.9	1.7	1.5	1.5
	MEAN	2.9	2.7	2.7	2.4	2.4

REMOVE COMBINED 927& 950 UNITS					
MAX	3.3	3.4	3.4	3.4	3.4
MIN	2.0	1.9	1.7	1.5	1.5
MEAN	2.6	2.5	2.5	2.3	2.4

Table 6—1 GSHP efficiencies, by boundary

ASHP

CODE	Comments	SPFH2	SPFhps	SPFH4	SPFH5	SEEF
418	SPFH2 based on ASHP only. Combined AS & EA No Circ Pump	2.4			1.7	1.7
422	ASHP + Thermal store	2.2			1.6	1.6
423	34% of work in (v16) missing	4.0		3.6	3.0	3.0
424	metered 6kW backup			2.3	2.1	2.1
425	EAHP				1.5	1.5
426	9% of work in (v16) missing	3.0		2.7	2.6	2.6
429	metered 5kW backup. v24 < v21, faulty off HP metering			2.2		2.2
440	unmetered desuperheater		2.1		1.6	1.6
441	unmetered desuperheater		2.1		2.2	2.2
442			2.3	2.3		2.3
443			2.4	2.4		2.4
444			1.7	1.7		1.7
445			2.6	2.6		2.6
446			2.5	2.5		2.5
447	metered backup w/o HP output			1.7		1.5
472	unmetered 5kW backup			1.2		1.2
473	23% v24 missing				1.6	1.6
474	metered backup			2.0		2.0
475	metered backup			1.7		1.7
478	looks like UVHWS			1.7		1.7
479	looks like UVHWS			2.1		2.3
486	metered 3kW min				1.8	1.8
487	metered 3kW min				1.5	1.5
488	DHW backup plus 2ndry rtn circ				1.9	1.9
	MAX	4.0	2.6	3.6	3.0	3.0
	MIN	2.2	1.7	1.2	1.5	1.2
	MEAN	2.9	2.2	2.2	1.9	2.0

REMOVE 423					
MAX	3.0	2.6	2.7	2.6	2.6
MIN	2.2	1.5	1.2	1.5	1.2
MEAN	2.6	2.1	2.0	1.8	1.9

Table 6—2 ASHP Boundary efficiencies

Analysis of Tables 6-1 and 6-2 indicates few SPF_{H2} values. As noted earlier, SPF_{H2} is a most useful assessment of heat pump efficiency since it includes just the heat pump compressor and controls plus the source pump or fan and can be directly compared with other heating sources such as gas and oil boilers. Note that the highest SPF_{H2} values quoted in the DECC report for both ground and air source are subject to doubt because of incomplete data, and their removal lowers the trial means for both types of heat pump.

SPFhps, whilst formerly a recognised Swedish boundary, is now an anachronism having been superseded by the Sepemo boundary definitions and, more importantly, all SPFhps values also fall into the SPF_{H4} category, albeit with their inclusion within a backup heater boundary. For integrated backup units, not separately metered, there is no way of identifying the impact of resistance heating on the SPF and thus their classification as

SPFhps or SPF_{H4}. In the context of the UK trials, SPFhps can be assigned to SPF_{H4} with no loss of data since in all cases the values for SPFhps are equal to those for SPF_{H4}.

Whilst SPF_{H3} is the preferred boundary for the Fraunhofer trials, it is ignored in this analysis. Of the 74 trial heat pumps, only 12 of the 51 ground source have metered circulation pumps and none of the air source. The trial data suffers from both a variation in number of monitored days and the percentage of monitored data collected and therefore requires a normalising process to estimate the relatively low circulation pump annual energy demand. The normalised range is from 100 to 400 kWh/year, Table 6-3, compared to the UK SAP 2009 procedure where Table 5f identifies 130 kWh/yr for a central heating pump (DECC/BRE, 2009, p 184)

CODE	Metered Wh/yr	days	% of year	% data collected	normaliser	Normalised Wh/yr	Normalised kWh/yr
457	84568	360	0.99	0.99	0.98	86609	87
463	106301	360	0.99	0.99	0.98	108866	109
462	93981	298	0.82	0.99	0.81	116274	116
470	134272	364	1.00	1	1.00	134641	135
458	146451	363	0.99	0.99	0.98	148745	149
465	131029	303	0.83	1	0.83	157840	158
464	141419	303	0.83	1	0.83	170356	170
460	149772	279	0.76	1	0.76	195938	196
461	104675	219	0.60	0.72	0.43	242303	242
469	349507	353	0.97	0.97	0.94	372565	373
414	329468	314	0.86	1	0.86	382980	383

Table 6—3 Circulation pump annual energy

Note: [(% of year) x (% data collected) = normaliser]. [(Metered Wh/yr)/normaliser = (Normalised Wh/yr)]

A box plot provides a graphical view of the normalised energy, Figure 6-1. It is evident that the whilst the median is 158 kWh/yr, circulation pump energy demand can be more than double this value and supports a move away from fixed head to variable speed circulators. Statistically, outliers are traditionally defined by means of "h", where h = 1.5 x IQR (interquartile range) and the lower and upper inner fences (LIF and UIF)[10].

Based on the normalised data in Table 6-3:

$h = 1.5(Q3 – Q1) = 140$; $LIF = Q1 – h = (-47)$ and $UIF = Q3 + h = 234$.

Note that the three highest demands would be classed as outliers.

[10] http://www.stat.wmich.edu/s160/book/node8.html

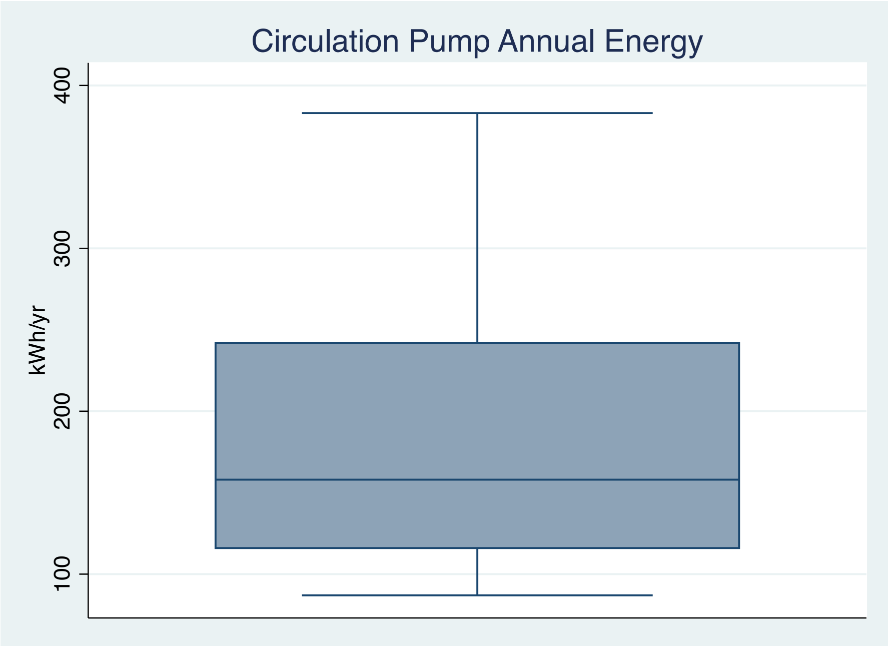


Figure 6—1 Circulation pump normalised annual energy

Given the limited number of SPF_{H2} systems, SPF_{H3} can only be calculated by subtraction of a circulation pump load from SPF_{H4}, leading to an SPF_{H3} value based on some form of annual circulator demand whether mean, median or mode, with or without outliers, and resulting in some instances with quite specious results where the value of SPF_{H3} is lower than SPF_{H4} and SPF_{H5}. For this reason, SPF_{H3} is ignored.

SPF_{H4} is useful since it represents the impact of heat pump, backup and circulation pump and thus the total system inputs and outputs. It is also the preferred Danish boundary.

SPF_{H5} is the most important boundary for the occupier since it is SPF_{H5} which best indicates the real running cost of the system. The efficiency at this boundary however, is also dependent on the heat losses from the hot water cylinder and therefore offers little in terms of heat pump efficiency analysis.

Finally, SEFF is redundant as a boundary definition since it conflates all systems whether or not there is backup heating or domestic hot water. All SEFF results are available as either SPF_{H4} or SPF_{H5} and therefore its results could be dropped without any loss of data. Unfortunately, due to anomalies in the design of the monitoring, SEFF remains the largest sample available under a single classification. SPF_{H5} applies only to a specific system design (measured hot water draw off), there are limited numbers of SPF_{H4} and fewer still of SPF_{H2} systems.

It is therefore apparent that the principle boundaries should be SPF_{H2}, SPF_{H4} and SPF_{H5}. Where performance values are available these boundaries will be the subject of this chapter, SEFF will be used where this is not the case.

Applying system taxonomies

The next twenty-odd pages present empirical evidence for a series of relationships between taxonomic features and performance. In most cases, there is some theoretical reason for expecting a relationship between the variables compared, however, there are also many potential reasons why the expected relationships might not be found in the data. Where expected relationships are not found, the author attempts to offer an explanation. It is therefore appropriate at this point to establish an overview that classifies systems by their component parts and the monitored data available, in order to investigate the general lack of consistent performance identified in previous chapter and provide a critical evaluation of the design of the EST trial.

Such an analysis of heat pump performance requires the formal assignment of system morphologies to establish taxonomical categories that may be utilised for the interrogation of same or similar groups and individual system performance. A Linnaean-type approach is suggested based on a hierarchy, which will contextualise system morphologies, including heat pump mechanical design, heat source, hydraulic system design and components, type of heat sink and ultimately control regime. It is expected that this classification will be of some use when allied with the various trial monitoring setups to shed light on the possible reasons for underperformance.

Any such classification is itself subject to arbitrary definitions since it proceeds from the choice of heat sources to provide thermal comfort and domestic hot water, a choice that compares the relative merits of gas, oil and electricity in terms of availability, price per kWh of heat output, carbon dioxide emissions, prime cost and life cycle cost. Electricity is supplied to all dwellings and may be utilised (with an exergy efficiency of the order of 5-20%) through resistance heating or more efficiently, by using heat pumps.

Heat pumps sit in two common categories of cycle design based on either the vapour compression cycle or the vapour absorption cycle. Vapour compression, the most common design, may be sub-divided into sub-critical HFC refrigerant cycles or trans-critical carbon dioxide cycles associated with the new breed of high temperature hot water heat pumps such as the Japanese 'Eco-cute' (Mitsubishi and Sanyo, online). The UK trials focus entirely on sub-critical vapour compression models. Thermodynamic work into the system may be by fixed or variable speed compression, which in theory, should lead to a closer matching of system output to demand and thus higher efficiency. The UK trials provide only three units with variable speed compression, with two from one manufacturer; all other units have fixed speed compressors. Backup heaters providing bi-valent operation may support cycle work-in.

The installation taxonomy begins with classification by heat source whether ground, water or air. This study has identified 51 "ground source heat pumps". There are three water source (an "open source artesian spring" and two "slinky under pond") which, for the purposes of classification, join the ground source grouping since shallow water temperature will have a smaller annual variation compared to outside air. There are 24 "air source heat pumps" with one exhaust air heat pump and one combined installation of air and exhaust air heat pump. The three air to air systems which were included in the trials have been ignored in all trial write-ups partly because of incomplete monitoring data and also because they are atypical of UK heat pump residential central heating.

Theoretically, a monovalent heat pump will have a higher seasonal efficiency than bivalent with electrical resistance backup. Since domestic hot water should be stored at or as near as possible to 60°C, a system of space heating and hot water should, in theory, be less efficient than one with lower temperature space heating only. Finally, the emitter type will impact on space heating water temperature where underfloor heating has the lowest temperature, followed by mixed underfloor and radiators and finally radiators only. However, a mixed system will require space heating flow at the higher radiator temperature with mixer valve temperature reduction to the underfloor heating thus negating some of the theoretically higher efficiency.

A taxonomic analysis provides the basis for a hierarchy of theoretical efficiencies by source. The basic taxonomy developed for analysis of all heat pump systems in the EST trial is shown in Figure 6-2 with twelve (12) potential system types based on mono or bivalent, space heating only or space heating and domestic hot water and finally the three emitter types underfloor, mixed and radiators. For both ground and air source, the data is then provided in terms of these taxonomical classes, the efficiencies at each class and boundary and finally a detailed breakdown of each system as evidence for the basic classification but also to provide additional data on the micro-differences between systems such as ground loop length, integrated circulation pump, buffer vessel, etc.

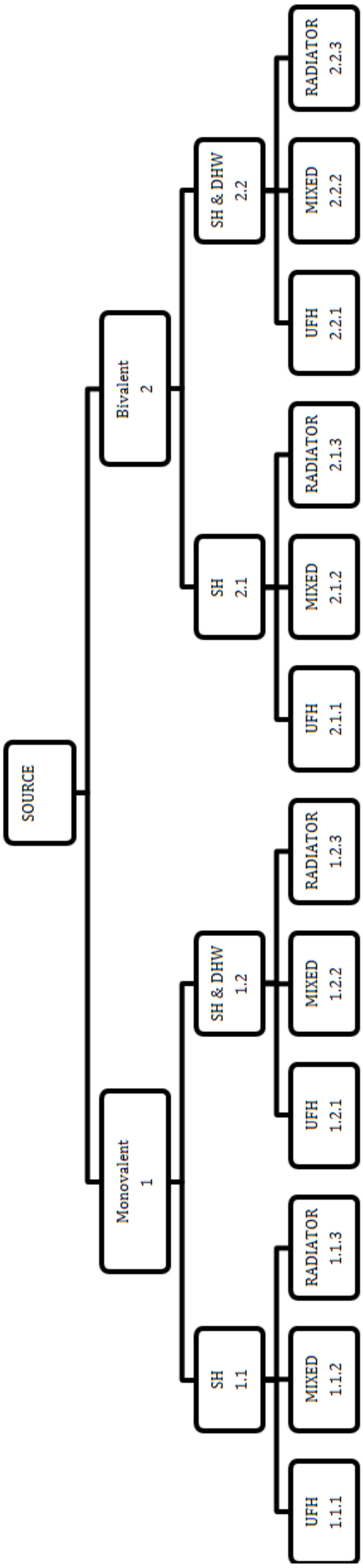


Figure 6—2 Taxonomical Classification for all heat pumps

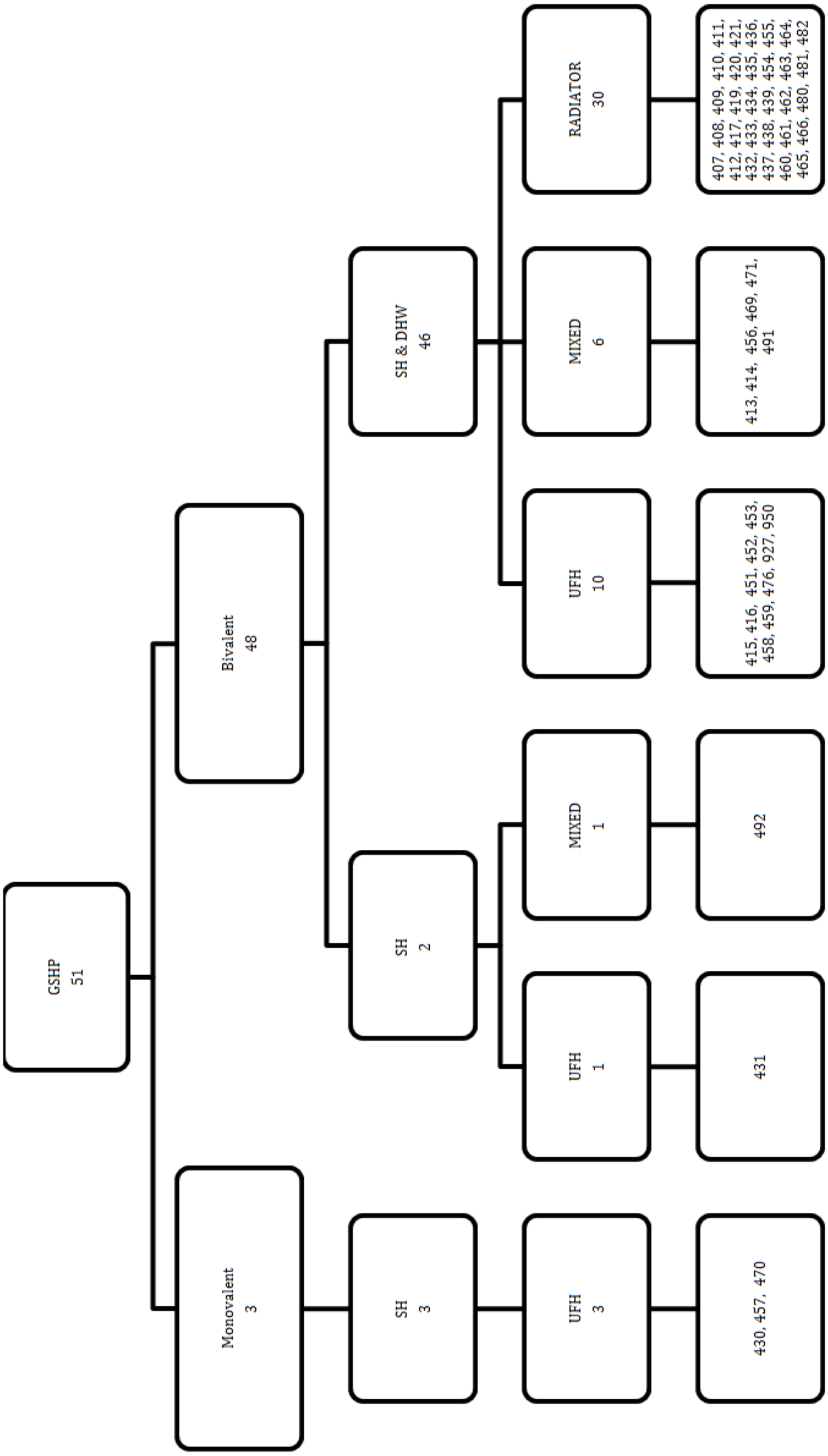


Figure 6—3 Ground Source Heat Pumps by taxonomy, with field trial ID Codes. Note that the bulk of the data is in taxonomy class 2.2., bivalent with space heating and DHW, with 30 radiator, 10 underfloor and 6 mixed emitter systems.

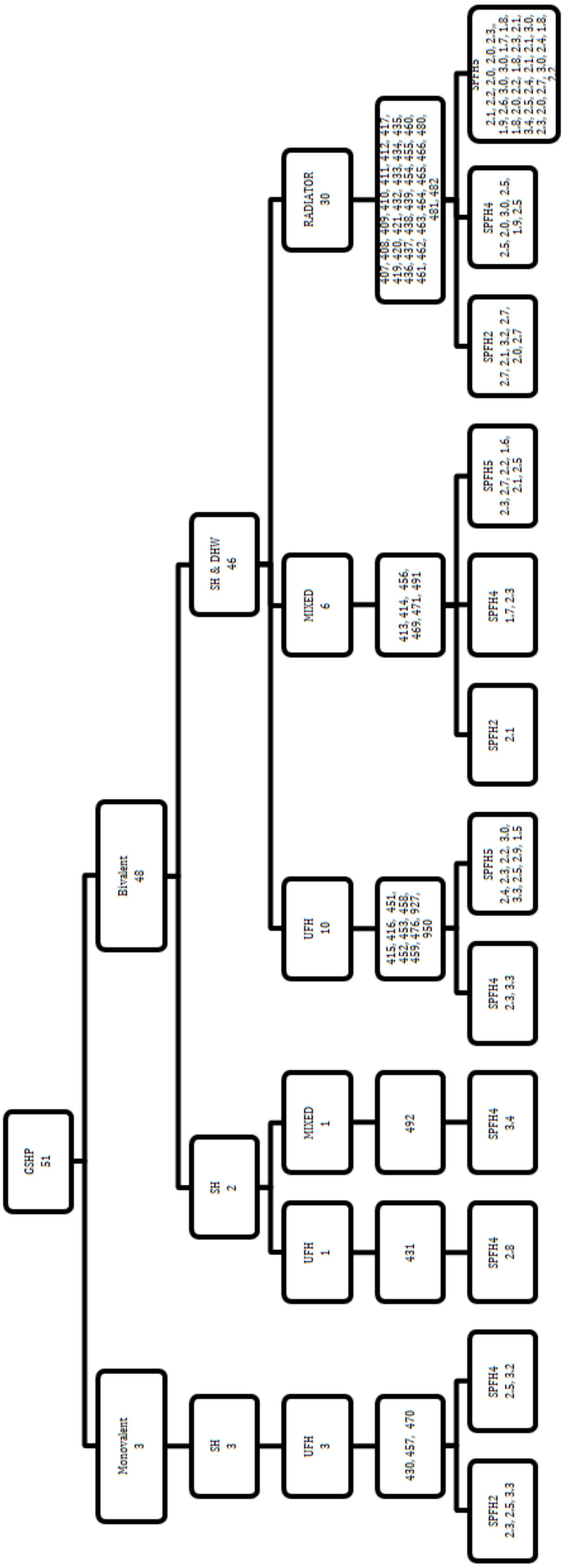


Figure 6—4 GSHP Taxonomy, ID Code, Boundary and SPF

GSHP	YES = 1 NO = 0				SOURCE		Heat pump													Comments			
CODE	Make	TYPE	kW	HP Input kW	Source	Source dimensions	Grnd Pump	Circulator	Boost	Integrated Buffer	Integrated DHW	Rads	UFH	Rads/UFH	Ext DHW	Immersion	Ext Buffer	Ext Circ pump	Ext Boost		SPFH2	SPFH4	SPFH5
407	Dimplex	2.2.3 BI/SHDHW/RAD	5	1.3	2 Borehole	80	1	1	0	0	0	1	0	0	1	1	0	0	0	oversized rads			2.1
408	Dimplex	2.2.3 BI/SHDHW/RAD	5	1.3	2 Borehole	80	1	1	0	0	0	1	0	0	1	1	0	0	0	oversized rads			2.2
409	Dimplex	2.2.3 BI/SHDHW/RAD	5	1.3	2 Borehole	80	1	1	0	0	0	1	0	0	1	1	0	0	0	oversized rads			2.0
410	Dimplex	2.2.3 BI/SHDHW/RAD	5	1.3	2 Borehole	80	1	1	0	0	0	1	0	0	1	1	0	0	0	oversized rads			2.0
411	Dimplex	2.2.3 BI/SHDHW/RAD	5	1.3	2 Borehole	80	1	1	0	1	0	1	0	0	1	1	0	0	0	oversized rads			2.3
412	Dimplex	2.2.3 BI/SHDHW/RAD	5	1.3	2 Borehole	80	1	1	0	1	0	1	0	0	1	1	0	0	0	oversized rads			1.9
413	Thermia	2.2.2 BI/SHDHW/MIX	6	1.8	3 Slinky (2 x 200m)	400	1	1	1	0	0	0	0	1	1	0	0	0	0				2.3
414	Thermia	2.2.2A BI/SHDHW/MIX/POOL	10	3	1 Spring		1	0	1	0	1	0	0	1	0	0	0	1	0	WSHP Pool			2.7
415	Thermia	2.2.1 BI/SHDHW/UFH	11	3.4	3 Slinky (3 x 200m)	600	1	1	1	0	0	0	1	0	1	0	0	0	0				2.4
416	Nibe	2.2.1 BI/SHDHW/UFH	5	1.2	4 Straight	300	1	1	1	0	1	0	1	0	0	0	0	0	0				2.3
417	Nibe	2.2.3 BI/SHDHW/RAD	12	3.6	3 Slinky	650	1	1	1	0	1	1	0	0	0	0	0	0	0				2.6
419	Nibe	2.2.3 BI/SHDHW/RAD	12	3.6	2 Borehole (2 x BH: 90 + 60m)	150	1	1	1	0	1	1	0	0	0	0	0	0	0				3.0
420	Nibe	2.2.3 BI/SHDHW/RAD	8	2.5	2 Borehole		1	1	1	0	1	1	0	0	0	0	0	0	0				3.0
421	Nibe	2.2.3 BI/SHDHW/RAD	8	2.5	4 Straight	600	1	1	1	0	1	1	0	0	0	0	0	0	0				1.7
430	Kensa	1.1.1 MO/SH/UFH	8		3 Slinky (2 x 40m)	80	1	1	0	0	0	0	1	0	0	0	0	0	0			2.3	
431	Dimplex	2.1.1 BI/SH/UFH	9	2.32	3 Slinky		1	1	1	0	0	0	1	0	0	0	1	0	0			2.8	
432	IVT	2.2.3 BI/SHDHW/RAD	4	<1.3kW	5.1 Vertical & Air panels		1	1	1	0	1	1	0	0	0	0	0	0	0				1.8
433	IVT	2.2.3 BI/SHDHW/RAD	4	<1.3kW	5.2 Horizontal panel		1	1	1	0	1	1	0	0	0	0	0	0	0				1.8
434	IVT	2.2.3 BI/SHDHW/RAD	4	<1.3kW	5.1 Vertical panel		1	1	1	0	1	1	0	0	0	0	0	0	0				2.0
435	IVT	2.2.3 BI/SHDHW/RAD	4	<1.3kW	2 Borehole		1	1	1	0	1	1	0	0	0	0	0	0	0				2.2
436	IVT	2.2.3 BI/SHDHW/RAD	6	1.3kW	2 Borehole		1	1	1	0	1	1	0	0	0	0	0	0	0				1.8
437	IVT	2.2.3 BI/SHDHW/RAD	6	1.3kW	2 Borehole		1	1	1	0	1	1	0	0	0	0	0	0	0				2.3
438	IVT	2.2.3 BI/SHDHW/RAD	6	1.3kW	2 Borehole		1	1	1	0	1	1	0	0	0	0	0	0	0				2.1
439	IVT	2.2.3 BI/SHDHW/RAD	11	2.6	3 Slinky (3 x 50m)	150	1	1	1	0	1	1	0	0	0	0	0	0	0				3.4
451	Worcester	2.2.1 BI/SHDHW/UFH	6.55	2	3 Slinky	300	1	1	1	0	0	0	1	0	1	1	0	0	0			2.3	2.2
452	Worcester	2.2.1 BI/SHDHW/UFH	10.4	3	3 Slinky		1	1	1	0	0	0	1	0	1	1	0	0	0			3.3	3.0
453	Worcester	2.2.1 BI/SHDHW/UFH	6.55	2	1 Pond		1	0	1	0	1	0	1	0	0	0	SH	1	0				3.3
454	IVT	2.2.3 BI/SHDHW/RAD	10.8	2.6	3 Slinky		1	1	1	0	0	1	0	0	1	0	0	0	1	oil no input only output			2.5
455	IVT	2.2.3 BI/SHDHW/RAD	10.8	2.6	5 Panels		1	1	1	0	1	1	0	0	0	0	0	0	0				2.4
456	Thermia	2.2.2 BI/SHDHW/MIX	7.13	2	4 Straight	400	1	1	1	0	1	0	0	1	0	0	0	0	0	v18 shows input			2.2
457	Kensa	1.1.1 MO/SH/UFH	16		3 Slinky		1	0	0	0	0	0	1	0	0	0	0	1	0		2.5	2.5	
458	Thermia	2.2.1 BI/SHDHW/UFH	16	4.1	3 Slinky		1	1	1	0	1	0	1	0	0	0	0	0	0	Secondary circ			2.5
459	Thermia	2.2.1 BI/SHDHW/UFH	8.1	1.7	3 Slinky		1	1	1	0	1	0	1	0	0	0	0	0	0				2.9
460	Calorex	2.2.3 BI/SHDHW/RAD	3.5		2 Borehole		1	0	0	0	0	1	0	0	1	1	0	1	0		2.7	2.5	2.2
461	Calorex	2.2.3 BI/SHDHW/RAD	3.5		2 Borehole		1	0	0	0	0	1	0	0	1	1	0	1	0		2.7	2.5	2.1
462	Calorex	2.2.3 BI/SHDHW/RAD	3.5		2 Borehole		1	0	0	0	0	1	0	0	1	1	0	1	0		2.1	2.0	2.1
463	Calorex	2.2.3 BI/SHDHW/RAD	3.5		2 Borehole		1	0	0	0	0	1	0	0	1	1	0	1	0		3.2	3.0	3.0
464	Calorex	2.2.3 BI/SHDHW/RAD	3.5		2 Borehole		1	0	0	0	0	1	0	0	1	1	0	1	0		2.7	2.5	2.3
465	Calorex	2.2.3 BI/SHDHW/RAD	3.5		2 Borehole		1	0	0	0	0	1	0	0	1	1	0	1	0		2.0	1.9	2.0
466	Thermia	2.2.3 BI/SHDHW/RAD	10.2	3	2 Borehole		1	1	1	0	1	1	0	0	0	0	0	0	0				2.7
469	Water Furnace	2.2.2 BI/SHDHW/MIX	12		4 Straight		1	0	0	0	0	0	0	1	1	1	SH	1	0	Data complete	2.1	1.7	1.6
470	Kensa	1.1.1 MO/SH/UFH	16		3 Slinky		1	0	0	0	0	0	1	0	0	0	0	1	0		3.3	3.2	
471	Worcester	2.2.2 BI/SHDHW/MIX	10.4	3	2 Borehole		1	1	1	0	0	0	0	1	1	0	SH	0	0	4 months missing data		2.3	2.1
476	ERW	2.2.1 BI/SHDHW/UFH	11		3 Slinky		1	1	0	1	1	0	1	0	0	0	0	0	0				1.5
480	Dimplex	2.2.3 BI/SHDHW/RAD	12	3	1 Pond		1	1	0	0	0	1	0	0	1	1	SH	0	0				3.0
481	Dimplex	2.2.3 BI/SHDHW/RAD	6	1.8	2 Borehole		1	1	0	0	0	1	0	0	1	1	1	?	1	Boost to buffer			2.4
482	Dimplex	2.2.3 BI/SHDHW/RAD	8	2.5	2 Borehole		1	1	0	0	0	1	0	0	1	1	1	?	1	Boost to buffer			1.8
491	Worcester	2.2.2 BI/SHDHW/MIX	9	3	3 Slinky		1	1	1	0	0	0	0	1	1	0	SH	1	0				2.5
492	Nibe	2.1.2 BI/SH/MIX	8	2.5	3 Slinky		1	1	1	0	0	0	0	1	0	0	0	0	0	Independent DHW		3.4	
927	IDM	2.2.1 BI/SHDHW/UFH	12		3 Slinky (7 x 50m)	350	1	0	0	0	0	0	1	0	1	1	0	1	0	427+428 24% of work in (v16) missing	5.6	4.8	4.7
950	Dimplex	2.2.1 BI/SHDHW/UFH	6.4	2	3 Slinky		1	1	0	0	0	0	1	0	1	1	1	0	0	450+468 20% of 450 SH (v21) missing			1.5
																				MAX	5.6	4.8	4.7
																				MIN	2.0	1.7	1.5
																				MEAN	2.9	2.7	2.4
																				REMOVE COMBINED 927& 950 UNITS			
																				MAX	3.3	3.4	3.4
																				MIN	2.0	1.7	1.5
																				MEAN	2.6	2.5	2.3

Table 6—4 GSHP Evidence for taxonomonies based on boundary analyss

GSHP system boundary, efficiency and taxonomy

For ground source heat pumps only six of the twelve potential taxonomical categories are represented in the trial but the bulk of the data lies in just three of these, all within the bivalent space heating and hot water category, Figure 6-3. We may wish to consider the efficiency relationship between taxonomy and system boundary since it is clear that radiators will require a higher mean water temperature than underfloor heating and therefore provide a connection to Carnot theoretical efficiency. Unfortunately the sample sizes for all taxonomies other than bivalent space heating and domestic hot water in boundary SPF_{H5} are too small to provide any useful empirical comparison, Tables 6-4 and 6-5.

GSHP TAXONOMY	TOTAL	SPFH2	SPFH4	SPFH5
1.1.1 MO/SH/UFH	3	2	3	0
2.1.1 BI/SH/UFH	1	0	1	0
2.1.2 BI/SH/MIX	1	0	1	0
2.2.1 BI/SHDHW/UFH	10	1	3	10
2.2.2 BI/SHDHW/MIX	6	0	0	6
2.2.3 BI/SHDHW/RAD	30	6	6	30

Table 6—5 Number of GSHP systems at each taxonomy and boundary classification

Analysis of bivalent space heating and domestic hot water by SPF_{H5} provides a semblance of the theoretical relationship best expressed through the median than the mean, Table 6-6.

SPFH5: Bivalent SH & DHW		
UFH	MIXED	RADIATORS
2.4	2.3	2.1
2.3	2.2	2.2
2.2	1.6	2.0
3.0	2.1	2.0
3.3	2.5	2.3
2.5	2.7	1.9
2.9		2.6
1.5		3.0
1.5		3.0
		1.7
		1.8
		1.8
		2.0
		2.2
		1.8
		2.3
		2.1
		3.4
		2.5
		2.4
		2.2
		2.1
		2.1
		3.0
		2.3
		2.0
		2.7
		3.0
		2.4
		1.8
2.4	2.2	2.3
2.4	2.3	2.2

Table 6—6 GSHP Bivalent SH & DHW: SPFH5 relationship between efficiency and emitter (sample mean in blue, median in red)

There should also be a clear relationship between the different boundary efficiencies for each taxonomical class, however, due to the lack of boundary data, this is only evident for bivalent space heating and domestic hot water, Table 6-7.

2.2.3 BI/SHDHW/RAD		
SPFH2	SPFH4	SPFH5
		2.1
		2.2
		2.0
		2.0
		2.3
		1.9
		2.6
		3.0
		3.0
		1.7
		1.8
		1.8
		2.0
		2.2
		1.8
		2.3
		2.1
		3.4
		2.5
		2.4
2.7	2.5	2.2
2.7	2.5	2.1
2.1	2.0	2.1
3.2	3.0	3.0
2.7	2.5	2.3
2.0	1.9	2.0
		2.7
		3.0
		2.4
		1.8
2.5	2.4	2.3

Table 6—7 GSHP Bivalent SH & DHW: Comparison of sample mean (in blue) for bimodal space heating and domestic hot water with radiators

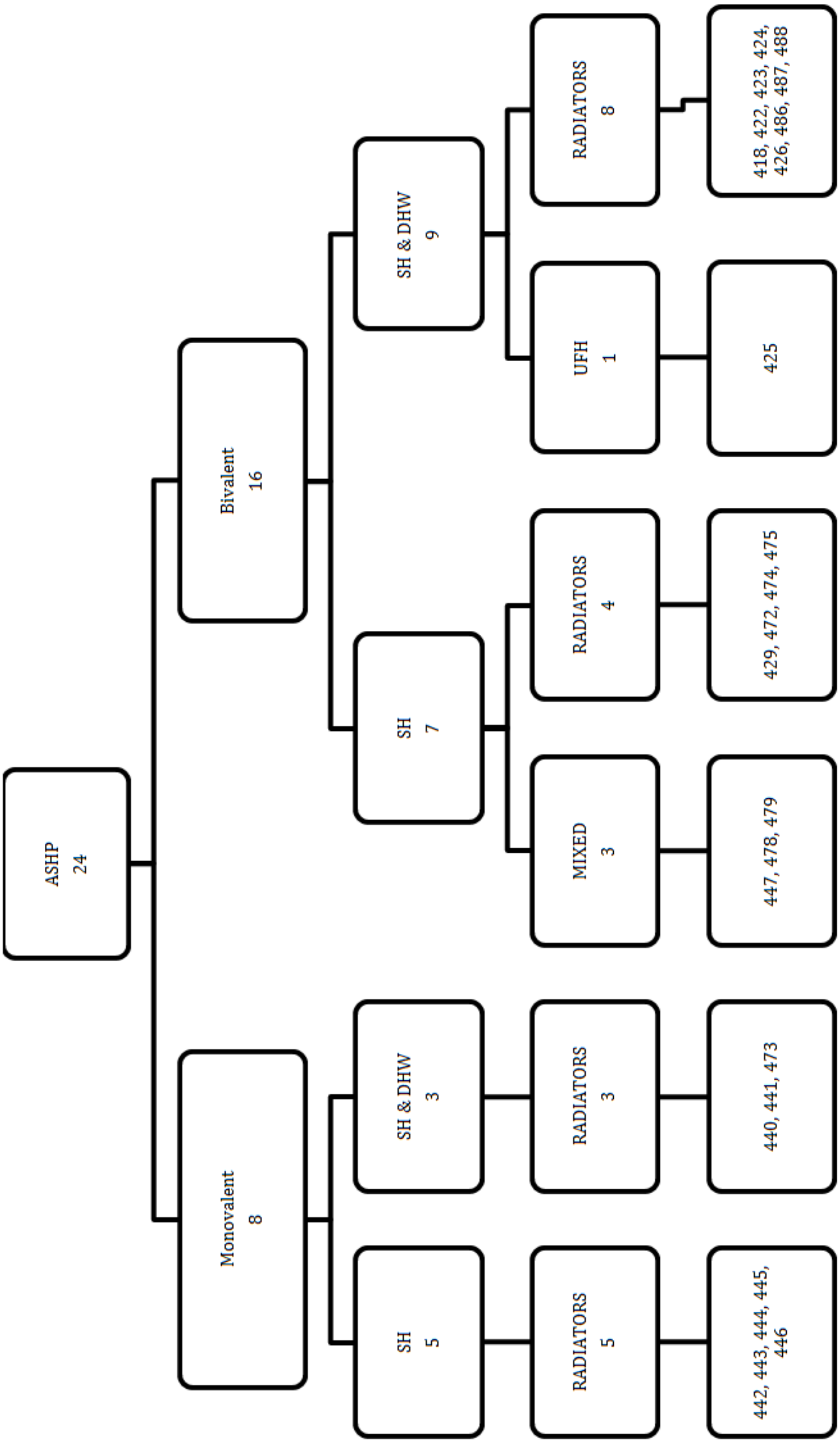


Figure 6—5 Air Source Heat Pumps by taxonomy and ID Code

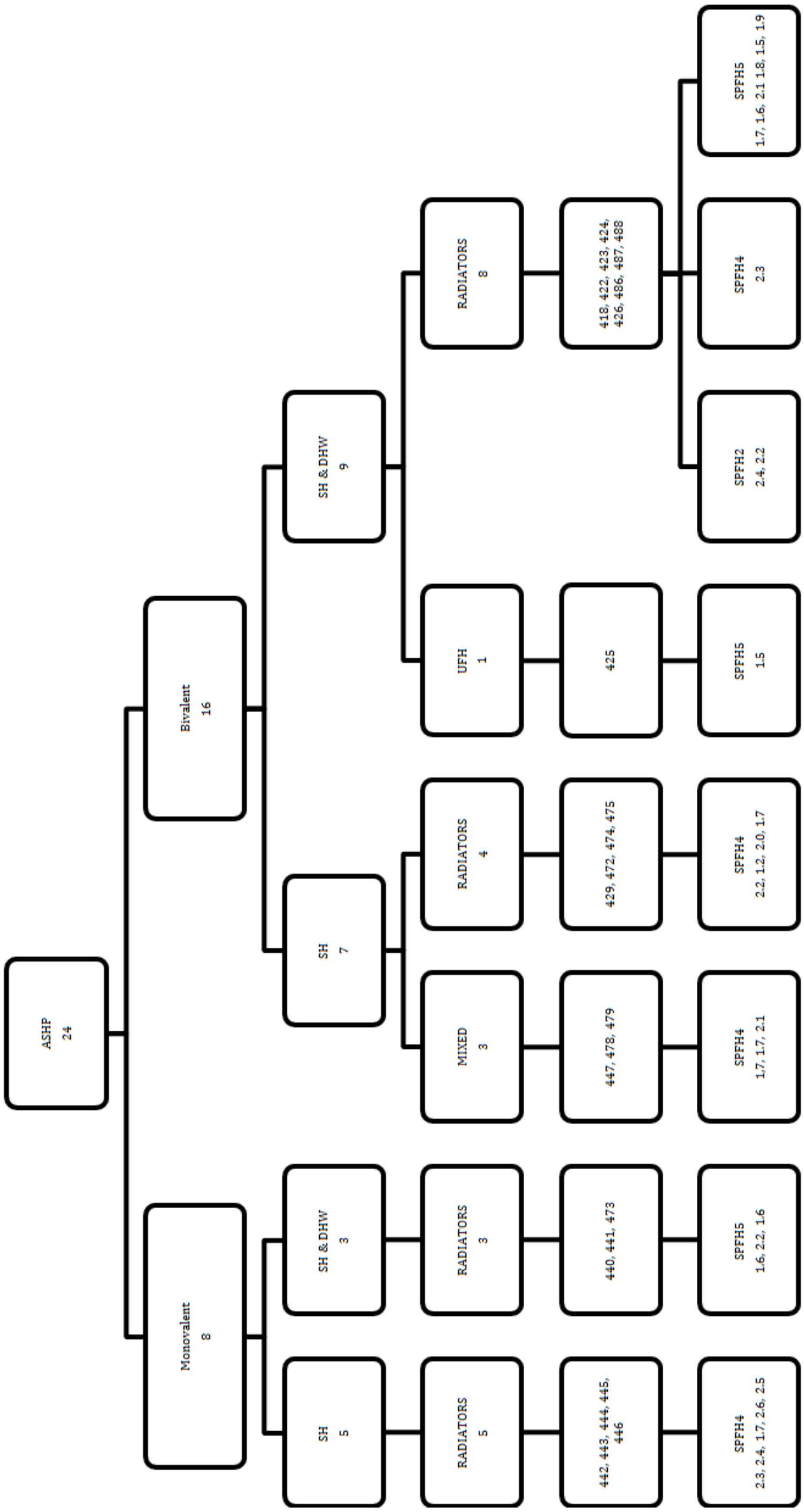


Figure 6—6 ASHP Taxonomy, ID Code, Boundary and SPF

ASHP	YES = 1 NO = 0				Type of ASHP		Within the Heat Pump															
CODE	Make	TYPE	kW	HP Input kW	External	Exhaust	Circulator	Boost	Integrated B	Integrated D	Rads	UFH	MIXED Rads/	Ext DHW	Immersion	Ext Buffer	Metered Ext	Ext Boost	Comments	SPFH2	SPFH4	SPFH5
442	Heat King	1.1.3 MO/SH/RAD	8.2	.	1	0	1	0	0	0	1	0	0	0	0	0	0	0			2.3	
443	Heat King	1.1.3 MO/SH/RAD	8.2	.	1	0	1	0	0	0	1	0	0	0	0	0	0	0			2.4	
444	Heat King	1.1.3 MO/SH/RAD	6.2	.	1	0	1	0	0	0	1	0	0	0	0	0	0	0			1.7	
445	Heat King	1.1.3 MO/SH/RAD	6.2	.	1	0	1	0	0	0	1	0	0	0	0	0	0	0			2.6	
446	Heat King	1.1.3 MO/SH/RAD	8	.	1	0	1	0	0	0	1	0	0	0	0	0	0	0			2.5	
440	Global	1.2.3 MO/SHDHW/RAD	9	.	1	0	1	0	0	1	1	0	0	0	0	SH	0	0	unmetered desuperheater			1.6
441	Global x 2	1.2.3 MO/SHDHW/RAD	18	.	1	0	1	0	0	1	1	0	0	0	0	SH	0	0	unmetered desuperheater			2.2
473	Ecodan	1.2.3 MO/SHDHW/RAD	8.5	.	1	0	1	0	0	0	1	0	0	1	1	0	0	0	23% v24 missing			1.6
447	Baxi	2.1.2 BI/SH/MIX	7.5	.	1	0	1	0	0	0	0	0	1	0	0	0	0	1	metered backup w/o heat out		1.7	
478	Worcester 1	2.1.2 BI/SH/MIX	11	0	1	0	1	0	0	0	0	0	1	0	1	0	0	1	looks like UVHWS		1.7	
479	Worcester	2.1.3 BI/SH/MIX	11	.	1	0	1	0	0	0	0	0	1	0	1	0	0	1	looks like UVHWS		2.1	
429	Baxi 9kW	2.1.3 BI/SH/RAD	9	.	1	0	1	0	0	0	1	0	0	0	0	SH	0	1	metered 5kW backup. v24 < v21		2.2	
472	Baxi 7.1kW	2.1.3 BI/SH/RAD	7.1	2.6kW	1	0	1	1	0	0	1	0	0	0	0	SH	0	0	unmetered 5kW backup		1.2	
474	Baxi 9.1kW	2.1.3 BI/SH/RAD	9.1	3.0kW	1	0	1	1	0	0	1	0	0	0	1	SH	0	1	metered backup		2.0	
475	Baxi 7.1kW	2.1.3 BI/SH/RAD	7.1	2.6kW	1	0	1	1	0	0	1	0	0	0	1	SH	0	1	metered backup		1.7	
425	Nibe 8kW	2.2.1 BI/SHDHW/UFH	7.4	2kW	0	1	1	1	1	1	0	1	0	0	0	0	0	0	EAHP			1.5
418	Nibe AS & Ex	2.2.3 BI/SHDHW/RAD	0	0	1	1	1	1	1	1	1	0	0	0	0	0	0	0	Combined AS & EA No Circ Pump	2.4		1.7
422	Nibe 8kW	2.2.3 BI/SHDHW/RAD	8	2kW	1	0	1	1	1	1	1	0	0	0	0	SH	0	0	ASHP + Thermal store	2.2		1.6
423	Nibe 8kW	2.2.3 BI/SHDHW/RAD	8	2kW	1	0	1	1	1	1	1	0	0	0	0	ALL	0	0	34% of work in (v16) missing	4.0	3.6	3.0
424	Dimplex 7.4k	2.2.3 BI/SHDHW/RAD	8	2.5kW	1	0	1	1	0	0	1	0	0	1	1	ALL	0	0	metered 6kW backup		2.3	2.1
426	Nibe 8kW	2.2.3 BI/SHDHW/RAD	8	2kW	1	0	1	1	1	1	1	0	0	0	0	0	0	0	9% of work in (v16) missing	3.0	2.7	2.6
486	Daikin 8kW	2.2.3 BI/SHDHW/RAD	8	0	1	0	1	0	0	0	1	0	0	1	1	0	0	1	metered 3kW min			1.8
487	Daikin 8kW	2.2.3 BI/SHDHW/RAD	8	0	1	0	1	0	0	0	1	0	0	1	1	0	0	1	metered 3kW min			1.5
488	Thermia Atri	2.2.3 BI/SHDHW/RAD	9.9	3kW	1	0	1	1	0	1	1	0	0	1	1	ALL	0	0	DHW backup plus 2ndry rtn circ			1.9
																			MAX	4.0	3.6	3.0
																			MIN	2.2	1.2	1.5
																			MEAN	2.9	2.2	1.9

REMOVE 423			
MAX	3.0	2.7	2.6
MIN	2.2	1.2	1.5
MEAN	2.6	2.0	1.8

Table 6—8 ASHP Evidence for Taxonomies

The air source heat pump taxonomic analysis is shown in Figures 6-5 and 6-6. The results of this breakdown by class is presented in Tables 6-8 and 6-9. Just six of the twelve possible system taxonomies arise but the range of system types is dominated by small sample sizes.

ASHP TAXONOMY	TOTAL	SPFH2	SPFH4	SPFH5
1.1.3 MO/SH/RAD	5	0	5	0
1.2.3 MO/SHDHW/RAD	3	0	0	3
2.1.2 BI/SH/MIX	3	0	3	0
2.1.3 BI/SH/RAD	4	0	4	0
2.2.1 BI/SHDHW/UFH	1	0	0	1
2.2.3 BI/SHDHW/RAD	8	4	3	8

Table 6—9 Number of ASHP systems at each taxonomy and boundary classification

There is little point in comparing the means for the different emitters since the largest classification is nine bivalent space heating with domestic hot water systems but eight of these are radiators.

It may be useful to compare the efficiencies across the boundaries for this classification, if only to note that the boundaries do exhibit the expected theoretical relationship even in such a small sample, Table 6-10.

2.2.3 BI/SHDHW/RAD		
SPFH2	SPFH4	SPFH5
2.4		1.7
2.2		1.6
	2.3	2.1
3.0	2.7	2.6
		1.8
		1.5
		1.9
2.6	2.5	1.9

Table 6—10 ASHP Bivalent SH & DHW: Comparison of sample mean (in blue) for bimodal space heating and domestic hot water with radiators

*** System 423 (Table 6-8) has 34% of the work-in electricity (v16) missing and is therefore omitted since any correction would be little better than a guess.**

Whilst the data provides some indication that such a taxonomy is useful, given the limited data sets, the most obvious reason why there may in fact be a poor correlation between space heating only and combined space heating and hot water, or, between different forms of emitter and overall system efficiency, is because actual system temperatures do not reflect the 'ideal model' assigned by the taxonomy. For example, only heat pumps with a change-over temperature control function can alternate between emitter and cylinder temperature requirements. For many of these radiator systems, the heat pump flow temperature is the same whether for space heating or domestic hot water.

Where mixed emitters are installed, it is most likely that the mean water temperature is the same for both radiators and underfloor heating unless a 3 port valve is installed to mix or divert between flow and return. However, under these circumstances, the heat pump would still be generating heat at the higher temperature and thus the lowest efficiency. A number of manufacturers suggest that even where weather compensation control is available, the heat pump is permanently switched to fixed temperature when supplying both space heating and domestic hot water. It would appear that there are practical challenges to adequate system temperature control when there are mixed emitters and both space heating and domestic hot water.

Mono or bivalent system design

At the time when the EST heat pumps were installed, there were no design rules in place that demanded the heat pump deal with 100% of the heating load and as a result, many of the systems have electrical resistance backup which supplies the base load. The trial therefore comprises both monovalent and bivalent space heating and, as is typical with UK installation, electrical immersion heaters support many of the hot water cylinders, thus also providing bivalent DHW heating. Bivalent heat pumps are often packaged units with internal resistance backup, presenting practical problems for field trial monitoring where separate monitoring of the resistance heater is either difficult within the small footprint of the unit, or where its introduction would potentially negate any warranty.

A complicating factor for the taxonomy is that all systems with a backup or boost function are classed as bivalent even though it is only possible to measure the full output of some of them. For electrical resistance heaters, system boundary analysis is dependent on the location of the backup heater in relation to monitoring heat meters. Efficiency at SPF_{H3} and SPF_{H4} includes any resistance heaters. For systems with integrated resistance heaters, heat meters are generally found downstream of the resistance heater and thus SPF_{H2} is not available. For systems with immersion heaters in separate cylinders, cylinder heat output, measured at the draw off hot water, will not represent cylinder heat input due to cylinder heat losses. The difference in efficiency due to system layout can be considerable with many cylinders in the EST field trials showing very high heat losses resulting from low hot water consumption or high immersion use.

The taxonomy should therefore be extended to differentiate between those systems with integrated electrical backup and those with immersion heaters in separate hot water cylinders. For the latter, a further division needs to identify those with heat metering of the cylinder primaries (the flow and return to the cylinder heat exchanger coil), that is SPFhps, SPF_{H3} or SPF_{H4}, from those where monitoring can only measure the hot water draw off, SPF_{H5}. Unfortunately, none of those systems with integrated DHW and few of those with separate cylinders are monitored to enable this comparison.

The difference between monovalent systems with space heating only or with space heating and domestic hot water must, in a properly functioning system, only be the use of higher temperatures to supply the domestic hot water. This could be explored through analysis of the space heating flow and hot water draw off temperatures but due to heat meter positions there are few examples where sufficient data is available. The best examples would be in classification band 1.2.1, monovalent with underfloor space heating and domestic hot water, however, there are no such examples in either the ground or air source data.

Of the 10 bivalent systems with space heating only, type 2.2.1, two are missing more than 20% of the data and seven have integrated boost. Only two of these integrated boost units provide SPF_{H4} where the default domestic hot water temperature is 52°C and a mixing valve is provided in conjunction with weather compensation for space heating. There is therefore no clear distinction in the measured data between temperatures for each heating system function.

Underfloor, mixed and radiators

Analysis by mean SPF_{H4} for ground source emitters only, provides a similar pattern of emitter efficiency as seen in the Danish trials (Pedersen, Jacobsen, 2011 p11), Figures 6-7, and thus provides some credibility to the taxonomic analysis.

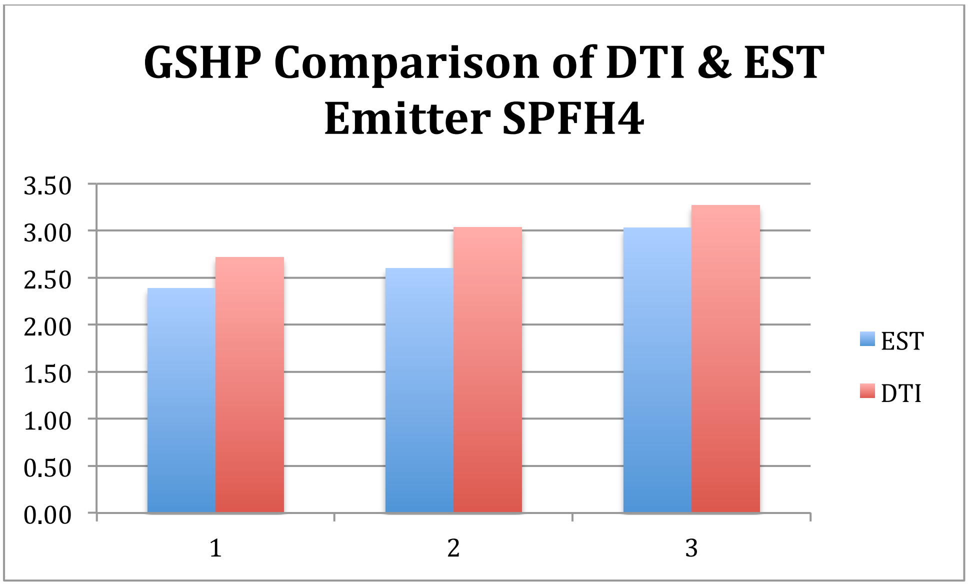


Figure 6—7 GSHP SPF_{H4} Efficiency Taxonomy by Radiator (1), Mixed (2), Underfloor heating (3)

Figure 6-8 for air source heat pumps is provided for SEFF since there are no underfloor heating systems measurable under SPF_{H4}, however, the graph does not reflect the expected relationship between emitter type and efficiency, perhaps due to sample size where only one unit has underfloor heating.

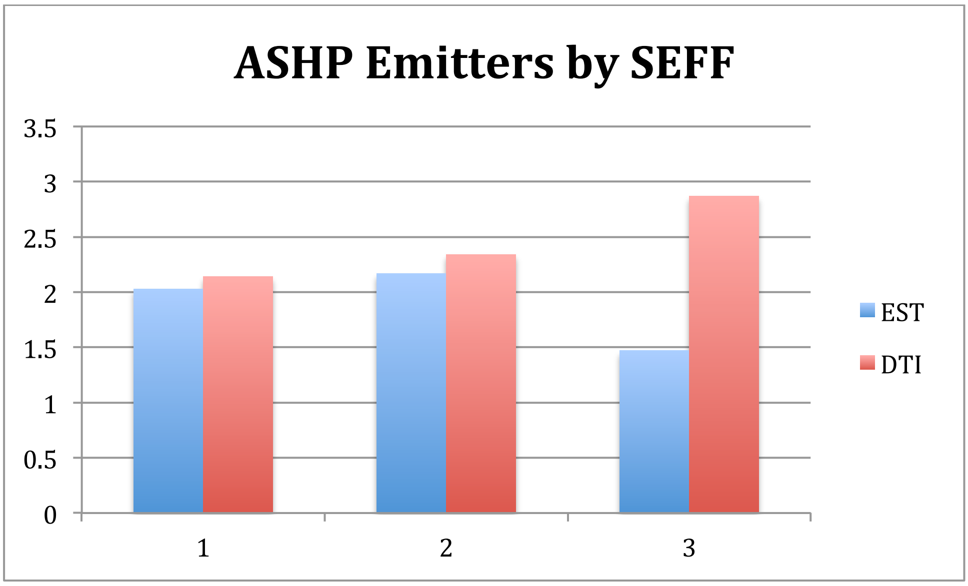


Figure 6—8 ASHP SEFF Efficiency Taxonomy by Radiator (1), Mixed (2), Underfloor heating (3)

Some remarks on taxonomy

The taxonomical analysis provides a structured overview of the individual systems that make up the trial population and identifies the limitations associated with a poorly defined methodological approach. The EST report (2010) aimed:

> "to determine how heat pumps perform in real-life conditions". It continues: "The sample was chosen to be broadly representative of the market at the time of commissioning the project, and included:
>
> - Air source and ground source heat pumps
> - Heat pumps installed in private and social housing properties
> - Heat pumps installed in new-build and retrofit installations
> - Heat pumps providing heating only
> - Heat pumps providing heating and hot water
> - Heat pumps installed with different heat delivery systems: under-floor heating and/or radiators
> - Grant-funded installations through LCBP and SCHRI
> - Systems combined with solar water heating"

Clearly these are very wide aims that require multidisciplinary, sociological and technological analyses. If the aims of the trial were to establish the seasonal performance of heat pump driven central heating systems as an alternative to solid fuel, gas, oil or resistance electrical heating, then the selection of case studies should have focused primarily on typical central heating systems with both space heating and domestic hot water driven by monovalent heat pumps and classified as Monovalent SH & DHW, type 1.2. If we consider this classification, none of the ground source, Figure 6-3, and only 3 (12.5%) air source, Figure 6-5, fall into this category. Thus the taxonomical analysis reveals that only 3 out of 75 (4%) of trial total fall into the most comparable classification to gas or oil central heating.

It should be recognised that it is not always possible to select monovalent heat pumps since many heat pumps, in principle, for good reasons, have a built-in resistance heater to support space heating during very cold weather and/or to support domestic hot water production and pasteurisation. Added to this category are those systems with a separate cylinder with immersion heater. For all bivalent systems, monitoring the specific output of any backup would provide direct comparison with the monovalent installations and help identify where and when backup heating is used. This Bivalent SH & DHW, type 2.2 classification, contains 46/51 (90%) ground source and 9/24 (37.5%) of the air source units.

Unfortunately a direct comparison between mono and bivalent units is only possible if resistance heating is monitored, hence the sub-division by monitoring protocol, Figures 6-4 and 6-6. Only SPF_{H2} and SPF_{H4} will provide direct evidence of heat pump operation because SPF_{H5} is compromised by its measurement of domestic hot water draw-off energy rather than the energy supplied to the cylinder whether integrated or separate. These unmeasured heat losses from the cylinder will ensure that the efficiency output numerator is under-valued in comparison to the input denominator. As for Seasonal Performance (SEFF), the chosen performance measure for the trial, since it combines space heating only systems and thus SPF_{H4}, with heating and hot water SPF_{H5} systems it further reduces the opportunity to analyse the specifics of design and performance.

The taxonomic analysis of ground source units provides only 7/51 in the SPF_{H2} and 10/51 in the SPF_{H4} categories, or only 17/51 (33%) at internationally recognised heat pump efficiency boundaries. For air source units the situation is worse with just 2/24 in the SPF_{H2} and 1/24 in SPF_{H4} classifications. This lack of a structured methodological trial design, results in the inability to clearly identify the role of emitter type and domestic hot water production on system efficiency.

Further analysis

In addition to the taxonomy of system design, it is possible to analyse some of the specifics of each installation including matching heat pump power to building design heat loss requirement, ground loop type and (where available) dimensions, role of buffer vessel, building time constant, emitter sizing and controls. These individual system sub-species provide further insight into the overall design of the EST trial and provide the data for a critique of individual heat pump system design consistency.

Matching heat pump output to RDSAP building heat loss

The sizing of a heat source for heating a building is primarily based on the design heat loss. The design heat loss is then subject to a fractional increase dependent on mode of operation (continuous or intermittent), thermal mass of the building plus the radiant and convective outputs of the emitters. Where domestic hot water is included, the standard UK allowance has traditionally been up to 3 kW. The CIBSE Domestic Heating Design Guide (CIBSE, 2011) establishes a plant energy factor of 15% greater than the building heat loss in order to account for preheating whilst confirming a range of 2 to 3 kW for domestic hot water. It is reasonable to assume that this is the case for "rule of thumb" heat loss calculations often encountered in domestic heating. We should expect to see heat pumps sized in the same way as gas boilers with a "plus 15 to 20%" rule. The EST Heat Pump Trial Site Report (EST, 2010) provides for each site a total floor area and an RDSAP value along with a building photograph. RDSAP, a "reduced" UK

SAP procedure, is based on a number of general assumptions about the building and its services and is used to provide Energy Performance Certificates (EPC) required when selling existing dwellings. EPCs for new dwellings are based on the full SAP procedure. With some judicious guesswork and RDSAP software[11], it is possible to reconstruct the RDSAP value and thus the heat loss coefficient (W/K). Although based on a number of generalisations, one would expect the results to reasonably model the building physics. With this in mind, it is possible to assess the heat pump sizing against building heat loss, Table 6-11.

ID	TFA	RDSAP	HLC	DHW 3kW	RDSAP LOAD kW	TOTAL LOAD	HP Output kW	HP % of Total Load	Without DHW load	COMMENTS
420	157	34	842	3	19.4	22.4	8	36%	41%	
454	330	37	933	3	21.5	24.5	10.8	44%	50%	oil backup
460	48	49	175	3	4.0	7.0	3.5	50%	87%	
461	48	57	155	3	3.6	6.6	3.5	53%	98%	SAP 53 with 300mm roof ins
463	70	70	143	3	3.3	6.3	3.5	56%	106%	SAP 60 as above
421	127	42	490	3	11.3	14.3	8	56%	71%	
423	98	29	481	3	11.1	14.1	8	57%	72%	
464	46	57	134	3	3.1	6.1	3.5	58%	114%	
465	46	57	134	3	3.1	6.1	3.5	58%	114%	end terrace?
462	70	64	124	3	2.9	5.9	3.5	60%	123%	
432	55	58	156	3	3.6	6.6	4	61%	111%	
433	55	58	156	3	3.6	6.6	4	61%	111%	
434	55	58	156	3	3.6	6.6	4	61%	111%	
435	55	58	156	3	3.6	6.6	4	61%	111%	
417	318	48	642	3	14.8	17.8	12	68%	81%	
436	98	53	250	3	5.8	8.8	6	69%	104%	
419	187	50	611	3	14.1	17.1	12	70%	85%	
437	102	56	236	3	5.4	8.4	6	71%	111%	
426	112	12	351	3	8.1	11.1	8	72%	99%	SAP 20 minimum
473	89	39	338	3	7.8	10.8	8.5	79%	109%	
455	259	55	453	3	10.4	13.4	11	82%	106%	
481	79	61	188	3	4.3	7.3	6.04	82%	140%	
416	226	77	132	3	3.0	6.0	5	83%	165%	SAP 67 maximum
407	48	60	130	3	3.0	6.0	5	83%	167%	
408	44	60	130	3	3.0	6.0	5	83%	167%	
409	46	60	130	3	3.0	6.0	5	83%	167%	
410	44	60	130	3	3.0	6.0	5	83%	167%	
411	45	60	130	3	3.0	6.0	5	83%	167%	
412	44	57	130	3	3.0	6.0	5	83%	167%	
480	183	48	439	3	10.1	13.1	12	92%	119%	
438	77.5	62	130	3	3.0	6.0	6	100%	201%	
482	80	57	213	3	4.9	7.9	8	101%	163%	
447	132	40	320		7.4	7.4	7.5	102%	102%	SAP 39 with booster
422	73	49	203	3	4.7	7.7	8	104%	171%	
429	127	32	371		8.5	8.5	9	105%	105%	
487	86	52	196	3	4.5	7.5	8	107%	177%	suspect SAP too low
418	251	63	355	3	8.2	11.2	11.9	107%	146%	
425	91	69	162	3	3.7	6.7	8	119%	215%	SAP 57 max
439	209	69	263	3	6.0	9.0	10.8	119%	179%	
486	68	57	153	3	3.5	6.5	8	123%	227%	suspect SAP too low
446	95	47	234		5.4	5.4	8.2	152%	152%	
445	48	42	174		4.0	4.0	6.2	155%	155%	
444	63	49	160		3.7	3.7	6.2	168%	168%	
442	98	59	208		4.8	4.8	8.2	171%	171%	
430	133	76	185		4.3	4.3	8	188%	188%	SAP 64 with v low u values
472	47	35	160		3.7	3.7	7.1	193%	193%	
474	70	46	186		4.3	4.3	9.1	213%	213%	
443	57	43	158		3.6	3.6	8.2	226%	226%	

Table 6—11 Correlation of Heat Pump output with RDSAP heat loss assessment

The kW output, Table 6-11, is based on the RDSAP heat loss coefficient (W/K) and a 23K design temperature difference, not untypical of UK dwellings. Where heat pumps are sized according to space heating load, a basic requirement for any heating design, they are over-sized for most of the heating season. Where output is automatically matched to heat load, as with variable speed compressors, this is less problematic than for fixed speed compressors where oversizing leads

[11] RDsapper available from: http://www.rusfa.com/RDsapper.htm

to cycling. One would expect there to be a close correlation between building design heat loss and the heat pump selected. Such a correlation is shown in Figure 6-9 where the relationship is limited to heat pump power being between 80% and 120% of building design heat loss, that is, where all outliers are removed and where the 'line of best fit' approximates y = x .

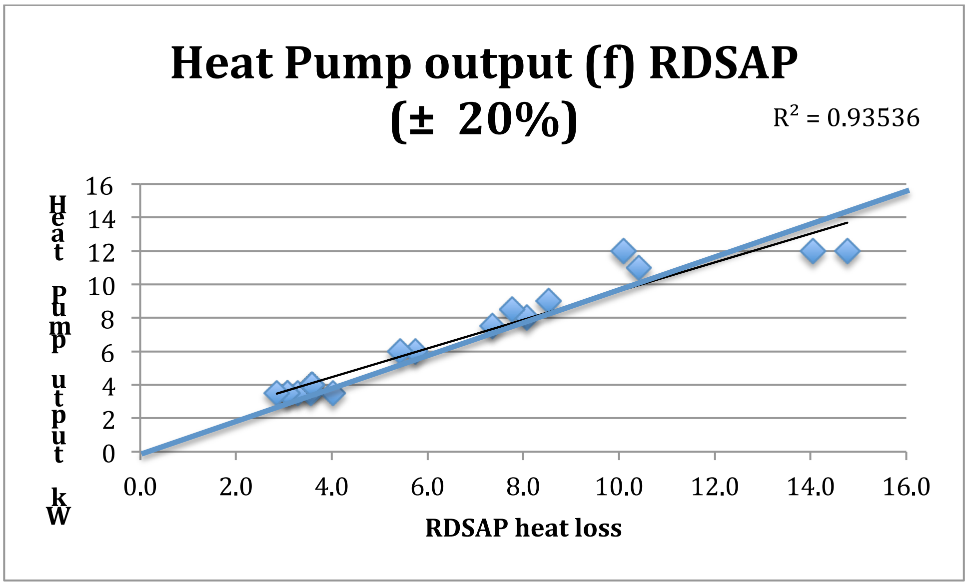


Figure 6—9 Heat pump output as a function of RDSAP heat loss for 80 to 120% (blue line represents ideal relationship, black line, 'line of best fit')

The relationship between heat loss and the heat pump chosen for the building would be expected to be reasonably linear where the designer first calculates the load and then provides a heat pump to supply this load. The trial provides a large number of systems with a significant mis-match between the heat pump selected and the building heat loss estimated by RDSAP, Figure 6-10, and whilst there is moderate correlation, the scatter suggests a degree of uncertainty as to the "correct" size of a domestic heat pump.

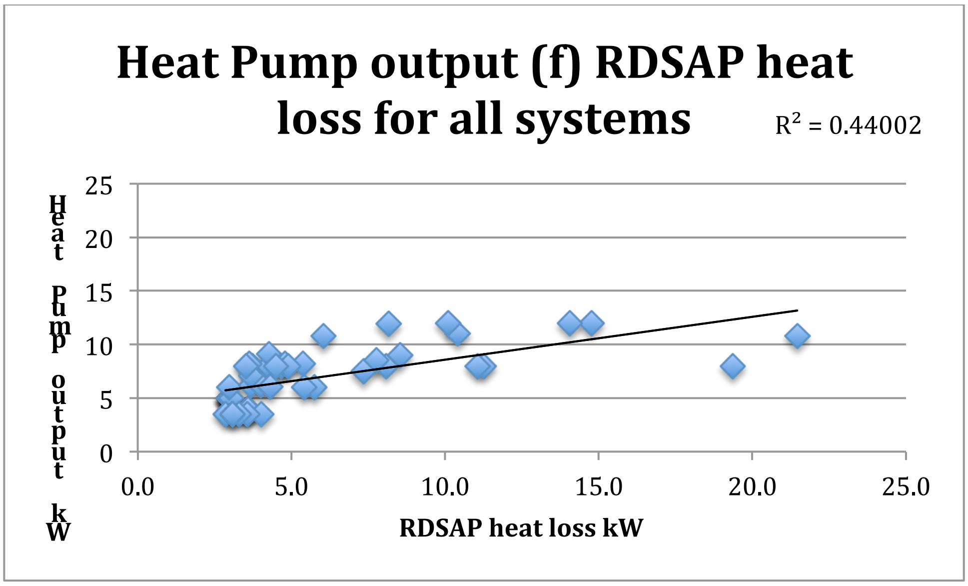


Figure 6—10 Heat pump output as a function of RDSAP for all systems

Where a system includes domestic hot water, let us assume an additional 3 kW allowance has been made, Figure 6-11. There is still considerable scatter confirming the degree of uncertainty identified in Figure 6-10.

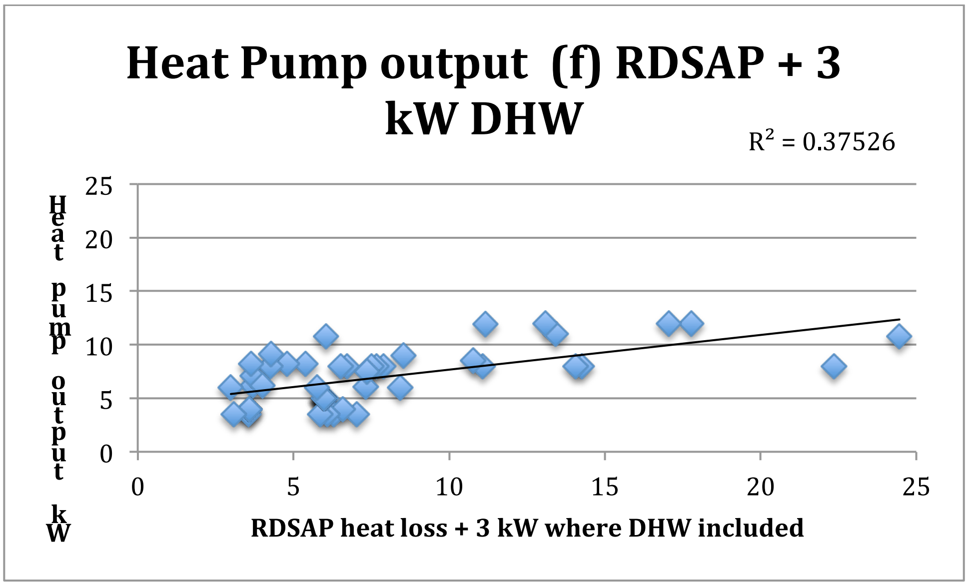


Figure 6—11 Heat pump output as a function of RDSAP plus 3 kW where there is a DHW load

Where a 3 kW allowance is made for domestic hot water, the sample shows a slight shift towards undersizing. Whilst undersizing would appear to be problematic, the need for a 3 kW domestic hot water allowance is questionable since hot water demand is dependent on numbers in occupation, whether occupants are at home all day, personal usage and size of DHW store. Annual dwelling hot water consumption in the trial ranges from less than 4,000 to greater than 80,000 litres per year or, on average, 11 to 220 litres per day. For the high end users, a

3kW allowance would provide daily hot water in a 4 hour period of heating. For well-insulated properties (with long time constants) and for low occupation levels (with low DHW use) it would appear that an additional hot water allowance might be unnecessary since the heat pump can safely operate bi-modally using traditional control systems which divert boiler flow to either space heating or hot water.

Ground loop type, length and efficiency

The trial heat pumps were installed before the advent of the Microgeneration scheme heat pump standard MIS 3005 in 2011 (the first edition no longer available), at a time when ground loops were likely to be designed by rule of thumb. Rules of thumb may be as vague as indicated by the following quotes:

> "For space heating, the figure presently used in the UK is 10 metres of horizontal Slinky trench for every 1 kW [100 W/m] of heat delivered from the heat pump, and for vertical systems, one 100 metre borehole should deliver between 3 and 5 kW [30 - 50W/m] of heat delivered from the borehole which means a single borehole is often sufficient for smaller properties" (Kensa Engineering manufacturer, online);

> "10 to 13m length per kW" [77 - 100 W/m], "62 W/m for a borehole" (Green Building forum, online);

and the author's favourite:

> "A (very rough) rule of thumb to calculate the ground loop area required, is twice the square footage of the building you are heating. The trench is around 1.8m deep where the ground temperature remains constant through the year." (Renenergy, designer and installer, online).

Whilst the EST trial provides information on all ground loop types (straight, slinky and borehole), in only 17 out of 52 installatons are ground loop dimensions given. No information is provided on the water source systems other than that 2 use slinkies as heat exchangers. There are 7 boreholes but 6 are the same length for the same heat pump in similar bungalows and so operate as a single sample.

Slinky ground loop lengths are taken from the confidential EST Heat Pump Trial Site Report (EST, 2010) and as such, are subject to scrutiny over their validity since a number of the reported slinky lengths are particularly short. Analysis of the coldest return temperature from the ground for these units provides little evidence of undersizing especially for loop lengths as short as 80 and 150 metres, if anything, there appears to be an inverse correlation which raises doubts as to the validity of the reported lengths, Table 6-12.

ID	Ground loop description	Length	Lowest return temp
417	Slinky 650m at a	650	-7.3
451	300m slinky in paddock	300	-2.5
413	Slinky 2 * 200m	400	-1.4
439	Slinky 3* 50m, 1m depth	150	-0.3
430	2*40m slinkies	80	0.4

Table 6—12 Slinky installation data

A slinky supplier, BHF Unlimited, provides the following: "This 50m slinky contains 300m of pipe. Requires a trench just 50m long" (BHF Unlimited). It is possible that at least some of the slinky length data may be quoted as trench length rather than actual pipe length. Further evidence of this possibility arises from manufacturers' installation sheets where, for example, Baxi provide the following (Baxi online):

> "A 50 metre slinky - made from 3 x 100 m length… A 40 metre slinky - made from 2.5 x 100 m lengths… A 30 metre slinky - made from 2 x 100 m lengths."

It is possible to recalculate some of the slinky lengths based on the assumption that what has been reported is trench length and to re-assess any correlation between heat pump output and length, Table 6-13. Although the sample is small, the adjustment of two of the reported lengths results in an improved correlation coefficient from 0.133 to 0.688, Figures 6-12 and 6-13.

ID	Ground loop description	Length	Lowest return temp	SPFH4	SPFH5	SEEF	kW output	Baxi length per slinky	Possible new total
413	Slinky 2 * 200m	400	-1.4		2.3	2.3	6.0		400
413	2*200m slinky	400			2.3	2.3	6.0		400
415	3*200m slinky	600			2.4	2.4	11.0		600
417	Slinky 650m at a depth of 1-1.5m.	650	-7.3		2.6	2.6	12.0		650
430	2*40m slinkies	80	0.4	2.3		2.3	8.0	250	500
439	Slinky 3* 50m	150	-0.3		3.4	3.4	11.0	300	900
451	300m slinky	300	-2.5	2.3	2.2	2.2	6.6		300

Table 6—13 Re-evaluating slinky length

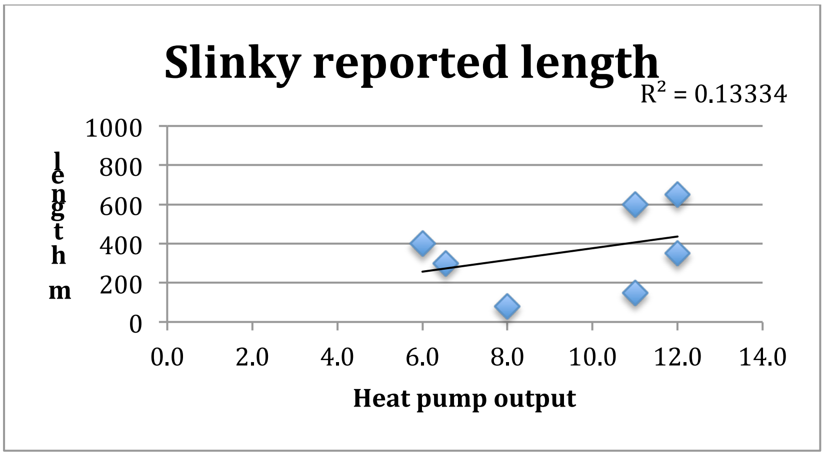


Figure 6—12 Reported slinky ground loop length as a function of heat pump output

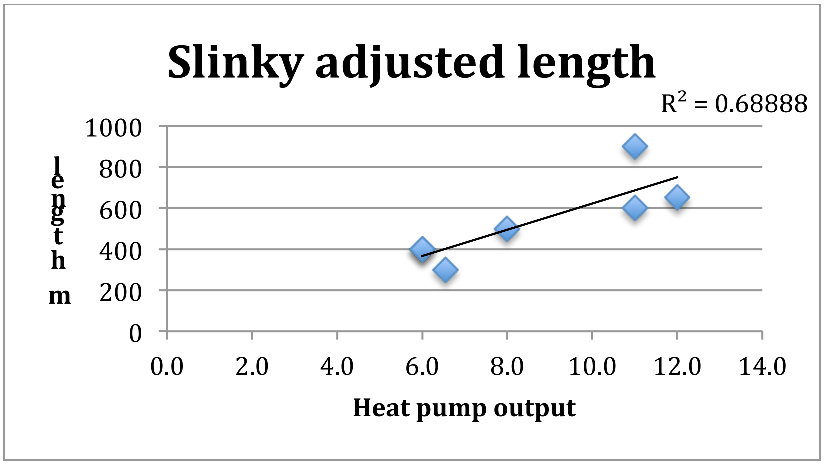


Figure 6—13 Adjusted slinky ground loop length as a function of heat pump output

The 3 straight ground loops do show a correlation but 3 is a very small sample for asserting that ground loop length is designed as a function of heat pump output, Figure 6-14.

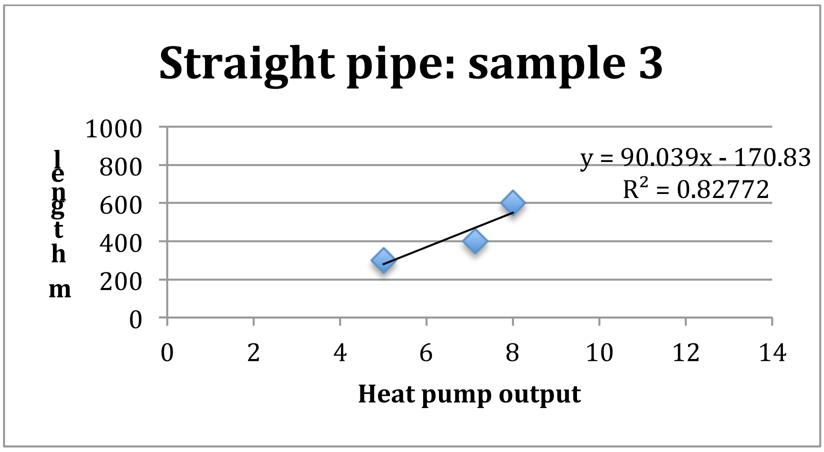


Figure 6—14 Straight ground loop length as a function of heat pump output

The greatest impediment to checking for correlation is the limitation of the trial monitoring protocol. There is no single measure of efficiency which can be use to assess all heat pump outputs, whether SPF_{H2}, SPFhps, SPF_{H3} or SPF_{H4}. The only metric that applies to all units is SEFF, for which we have already noted a number of drawbacks. However, given that SEFF is available for all ground source units, Figure 6-15, we note that the influence of ground source type is limited by the paucity of dimensioning information and the unevenness of the sample numbers.

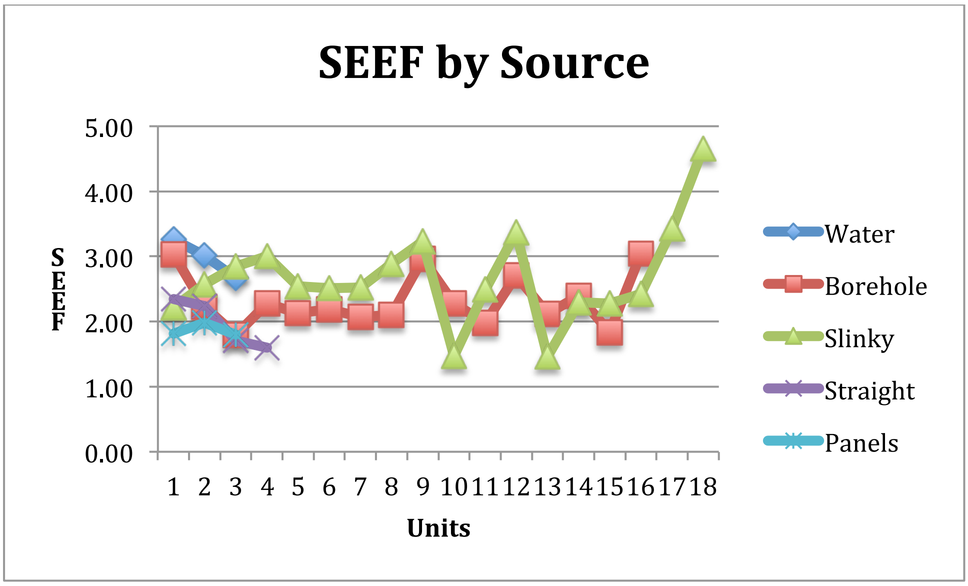


Figure 6—15 Ground loop type and SEFF

A smaller sample is provided by SPF_{H4}, Figure 6-16, where the SPF_{H4} borehole mean is 2.4 and the slinky 3.0. The Unit 9 slinky (ID Code 927) should be treated an outlier since it is missing 24% of the work-in data. If removed from the sample, the slinky mean reduces to 2.8, still well above that of the borehole sample.

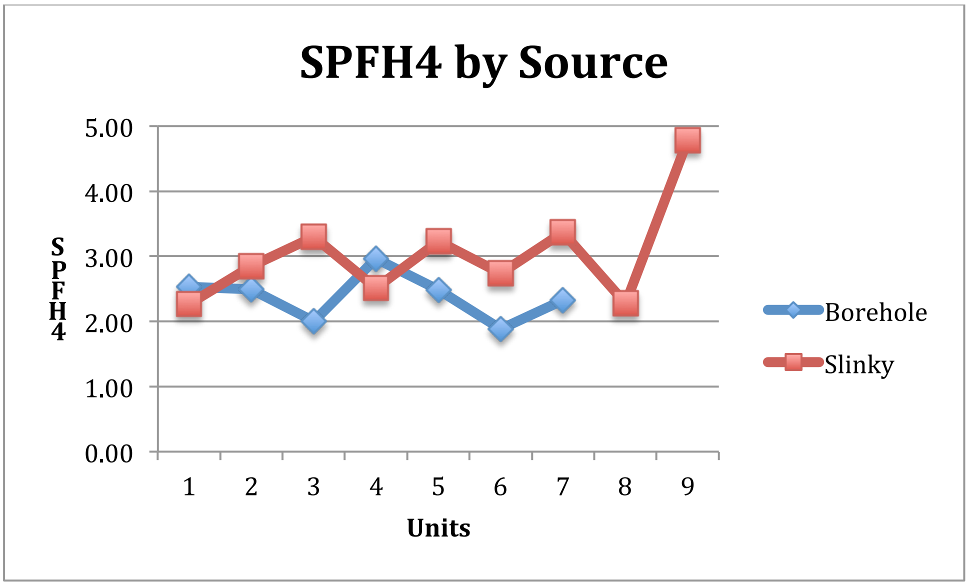


Figure 6—16 Comparison of slinky and borehole by SPF_{H4}

A ground source heat exchanger should be designed to supply the maximum heat output from the heat pump. Given that there will be differences in ground conductivities which will lead to variation in length, without the soil analysis from each site and an indication of water table height, it is not possible to assess whether there is any consistency in their design; there is certainly little consistency in the graphical analysis.

It would appear that there are at least three obstacles to assessing the efficacy of different sources; the sample size is too small for anything other than slinkies, there is insufficient length data for the different sources and finally, there is no single efficiency metric that can be applied to the whole sample which measures heat pump output only, that is, SPF_{H2}.

Correlation between buffer vessel and SPF

Traditional heat pump system design utilises a buffer vessel as a thermal store, its aim being to limit the number of compressor cycles and thus reduce standing losses, unnecessary starting surge amps and limit the inefficiencies associated with the vapour compression cycle needing to settle before providing any useful heat, “switching losses” (Griggs & McCall, 1988); it is thus a corrective to the oversizing phenomenon. Many of the functions of variable speed compressors duplicate those of the buffer vessel, in particular, variable speed should largely eliminate cycling.

Buffer vessel sizing is covered in EN 15450:2007, Heating systems in buildings — Design of heat pump heating systems:

> "In order to minimize cycling, it shall be assured that the heating capacity delivered by the heat pump is completely transferred to the heating system.
> NOTE: This can be achieved by setting a sufficient constant volume flow rate at the heat sink side of the heat pump. A higher inertia (capacity) can be achieved with a surface heating system or by installing a buffer storage (in parallel or series). A buffer storage connected in parallel with the heat pump serves additionally as a means of hydraulic decoupling. A guidance value for sizing the buffer storage volume is 12 to 35 l per kW maximum heat pump capacity."

Whilst EN 15450 provides a range of design options, both with and without buffer vessel, there is no specific guidance on whether a buffer vessel is necessary and where it 'should' be located in the system other than, it is suggested, in parallel with the space heating circuit. Where there is any design and installation guidance for a particular heat pump, it is provided by the heat pump manufacturer.

There appears to be little logic applied to the design criteria for buffer vessels in the EST trials. The trial shows three basic buffer vessel locations: the heat pump supplies a thermal store buffer vessel from which all loads are fed (a parallel hydraulic decoupling design) or the buffer vessel supplies either space heating only or domestic hot water only. Since the buffer vessel is designed to increase the efficiency of the system one might expect to see evidence for this in the trial data.

For the SPF_{H4} ground source heat pumps the sample number is 3 but one of these is missing 4 months data and therefore the number of viable buffer vessel systems is too small. For SEFF, there are 11 buffer vessel systems but at this boundary, the losses associated with hot water cylinders combined with immersion heater input, negate any useful conclusions being drawn, Figure 6-17. The situation is similar for air source systems, with SPF_{H4} providing almost the same mean with a buffer, 1.9, as without, 2.1, Figure 6-18. SEFF provides the same analysis with 1.8 versus 2.0 for with and without buffer, Figure 6-19.

Neither ground or air source heat pumps provide empirical evidence of the efficacy of installing buffer vessels, if anything, the evidence indicates higher SPF without buffer vessels.

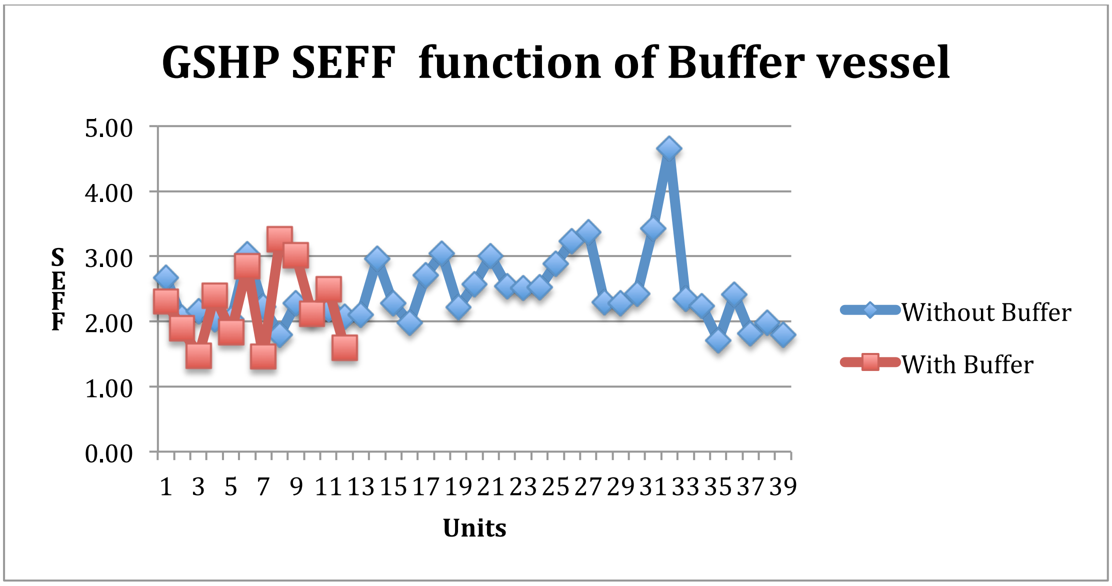


Figure 6—17 GSHP SEFF in relation to buffer vessel use

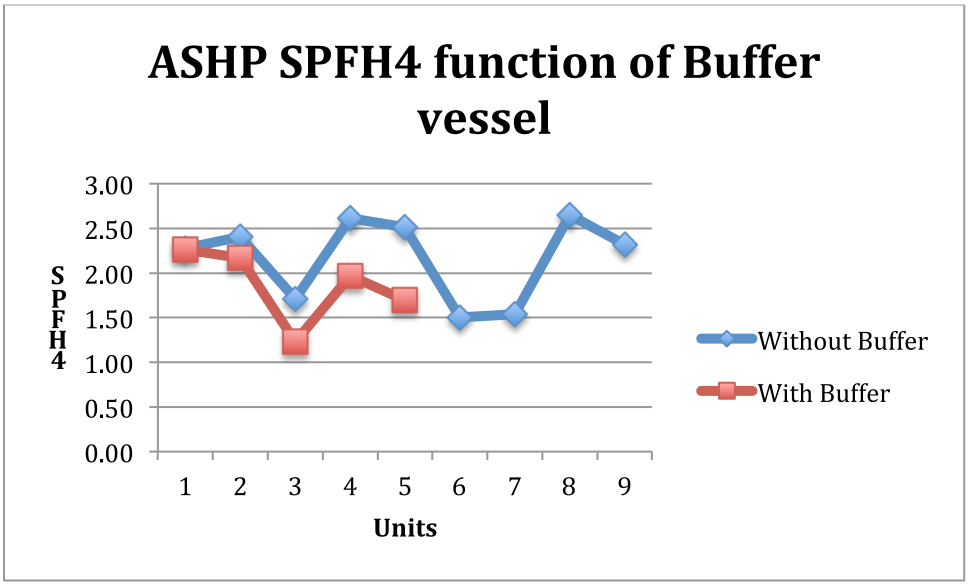


Figure 6—18 ASHP SPF_{H4} in relation to buffer vessel use

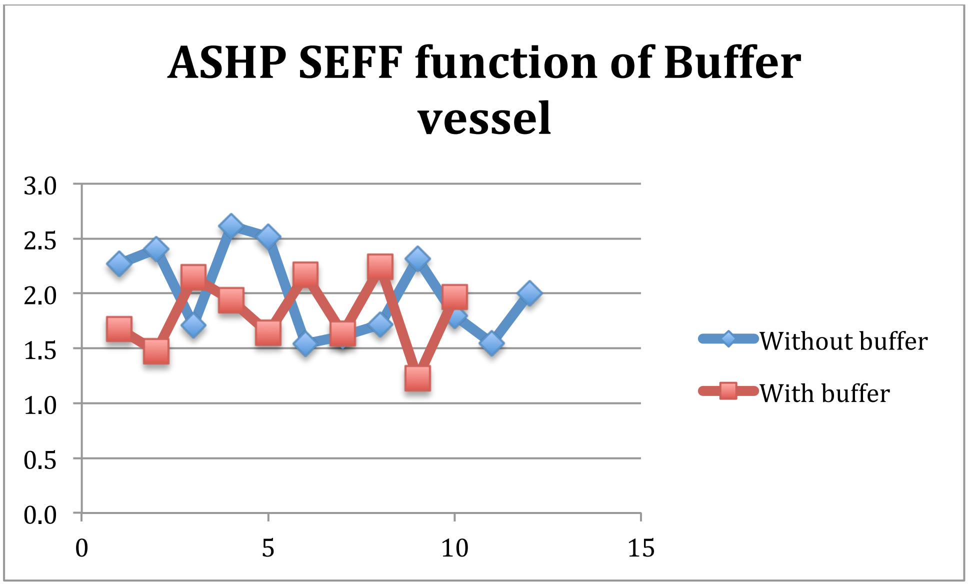


Figure 6—19 ASHP SEFF in relation to buffer vessel use

Similar results are reported from FAWA Swiss trials and Fraunhofer German trials:

> *"Technische Speicher haben keinen Einfluss auf die Jahresarbeitszahl"*, or, "buffer vessels have no effect on the seasonal performance factor." (Erb, et al, 2004 p82).
>
> "The most efficient [heat pumps] were those which charged the heating circuit directly, ie, systems without any buffer storage." (Miara, et al, 2011 p37)

The authors of both reports suggest that incorrect buffer sizing and heat losses may detract from any potential savings. Buffer vessel sizing must balance the need to provide sufficient water to reduce cycling whilst minimising vessel heat losses. It would appear that buffer vessels, like hot water cylinders need adequate levels of insulation, a fact acknowledged by some vessel manufacturers such as Gledhill: "the Buffer store is insulated to a very high standard", (Gledhill, 2013). Gledhill also provide a sizing guide, Figure 6-20, based, it is claimed, on EN 15450:

Heat Pump Sizing	
Cylinder capacity	**Heat pump output**
90 litres	2.6kW - 7.5kW
120 litres	3.4kW - 10kW
210 litres	6kW - 17.5kW
300 litres	8.5kW - 25kW
400 litres	11.4kW - 33.3kW

Figure 6—20 Buffer vessel sizing (Gledhill, 2013)

Thermal mass and the time constant

Since the heat pump will generally switch from space heating to hot water (DHW priority) whenever there is a call for DHW reheat, the impact on the space temperature will depend on the length of time the switch-over continues and building time constant. For typical mid to heavyweight buildings the time constant is generally long enough to retain sufficient heat, supported by any internal heat gains, for thermal comfort to be maintained.

The time constant (τ, tau) is the time taken for the building to cool to half its temperature, a thermal half-life. The time constant is dependent on thermal capacity and thermal transmission and, whilst calculations will necessarily be approximations, the time constant provides a time value that can be used to assess the impact of a changeover from space heating to hot water under design conditions. A series of such calculations based on light, medium and heavyweight structures for decreasing heat loss coefficient (W/K) is shown in Table 6-14.

In the examples the heat loss coefficient ranges from 350 down to 75 W/K, which for a 96m^2 building (8 x 6 x 5m) approximates a Passivhaus or UK Code Level 6 envelope design with a heat loss parameter of 0.8 W/m^{2}K.

Thermal mass and time constant							
Consider HLP = 0.8 W/m2K as practical construction limit							
96 m2 TFA							
Lightweight Timber frame							
HLC	**TFA**	**HLP**	**k**	**J/K**	**T1/2**	**s/h**	**τ1/2 (h)**
350	96	3.65	0.69	6,798,110	13,402	3,600	4
300	96	3.13	0.69	6,798,110	15,636	3,600	6
250	96	2.60	0.69	6,798,110	18,763	3,600	8
200	96	2.08	0.69	6,798,110	23,453	3,600	9
150	96	1.56	0.69	6,798,110	31,271	3,600	13
100	96	1.04	0.69	6,798,110	46,907	3,600	19
75	96	0.78	0.69	6,798,110	62,543	3,600	25
Medium Weight							
HLC	**TFA**	**HLP**	**k**	**J/K**	**T1/2**	**s/h**	**τ1/2 (h)**
350	96	3.65	0.69	15,244,800	30,054	3,600	8
300	96	3.13	0.69	15,244,800	35,063	3,600	14
250	96	2.60	0.69	15,244,800	42,076	3,600	17
200	96	2.08	0.69	15,244,800	52,595	3,600	21
150	96	1.56	0.69	15,244,800	70,126	3,600	28
100	96	1.04	0.69	15,244,800	105,189	3,600	42
75	96	0.78	0.69	15,244,800	140,252	3,600	56
Heavyweight							
HLC	**TFA**	**HLP**	**k**	**J/K**	**T1/2**	**s/h**	**τ1/2 (h)**
350	96	3.65	0.69	48,888,000	96,379	3600	27
300	96	3.13	0.69	48,888,000	112,442	3600	31
250	96	2.60	0.69	48,888,000	134,931	3600	37
200	96	2.08	0.69	48,888,000	168,664	3600	47
150	96	1.56	0.69	48,888,000	224,885	3600	62
100	96	1.04	0.69	48,888,000	337,327	3600	94
75	96	0.78	0.69	48,888,000	449,770	3600	125

Table 6—14 Time constant calculations, adapted from RJL/07/2010, UCL

For the smallest heat pumps with outputs of 3.5 kW, a UK traditional cylinder of 136 litres and a temperature rise of 45K (50 - 5), it would take about 120 minutes or 2 hours to heat the domestic hot water. For larger cylinders, say 250 - 300 litres, the time required is 4 hours. As can be seen from Figure 6-21, derived from Table 6-14, the impact on internal temperature of the changeover from space heating to domestic hot water will depend on the time constant.

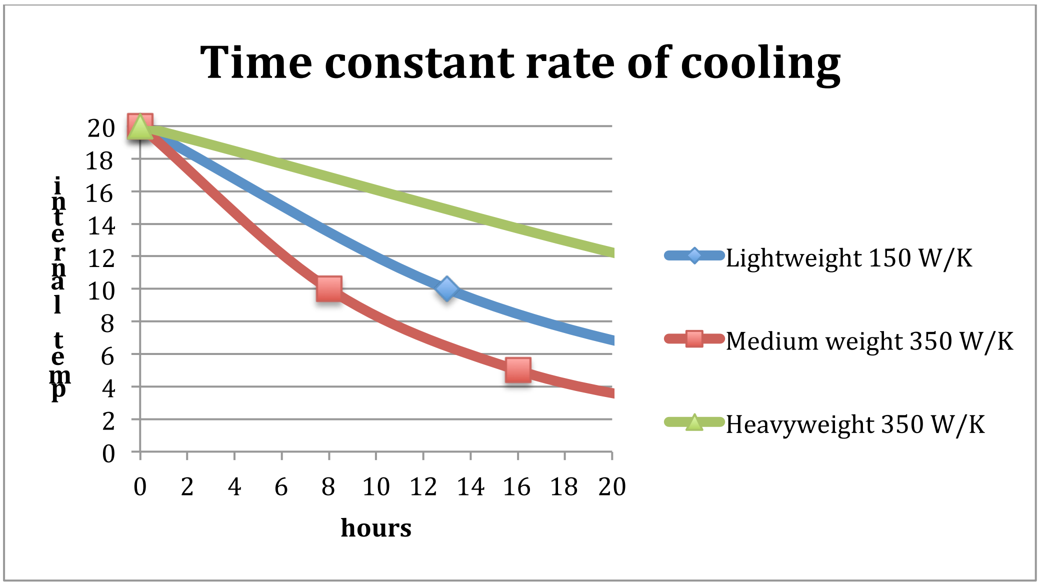


Figure 6—21 Time constant calculations

For a lightweight timber frame building with good levels of insulation, HLC = 150 W/K, the internal temperature would drop from 20 to about 18°C for the large cylinder and may provide an argument for the inclusion of a heat pump sizing hot water allowance, unless of course, DHW heat up is timed to occur over night when occupants are in bed. For poorly insulated and draughty medium-weight buildings, probably typical of existing UK housing stock, the fall in temperature is greater. For poorly insulated, heavyweight buildings, the theoretical temperature drop over a 2 to 3 hour period is small although draught will impact on comfort conditions, Figure 6-21. The obvious implication is to insulate and draught strip before considering a new heating installation. Where an allowance for DHW is made, the control system needs to be able to supply both loads simultaneously.

Efficiency and cycling

The design winter temperature difference is experienced in the UK for relatively short time periods, where mean winter temperature is generally about 7 to 11°C depending on location. This will mean that the heat pump output could be double the load for much of the year leading to rapid cycling and the attendant inefficiencies identified by other authors (Grigg & McCall, 1988, Stafford & Lilley, 2012).

The trials provide ample evidence of such cycling. Consider a stone barn, ID Code 470, converted in 2005, Figure 6-22, supplying underfloor heating over a two day period where outside temperatures range from about 6 to 16°C, Figure 6-23.

Figure 6—22 ID Code 470 Heavy weight building (EST Site Report, 2010)

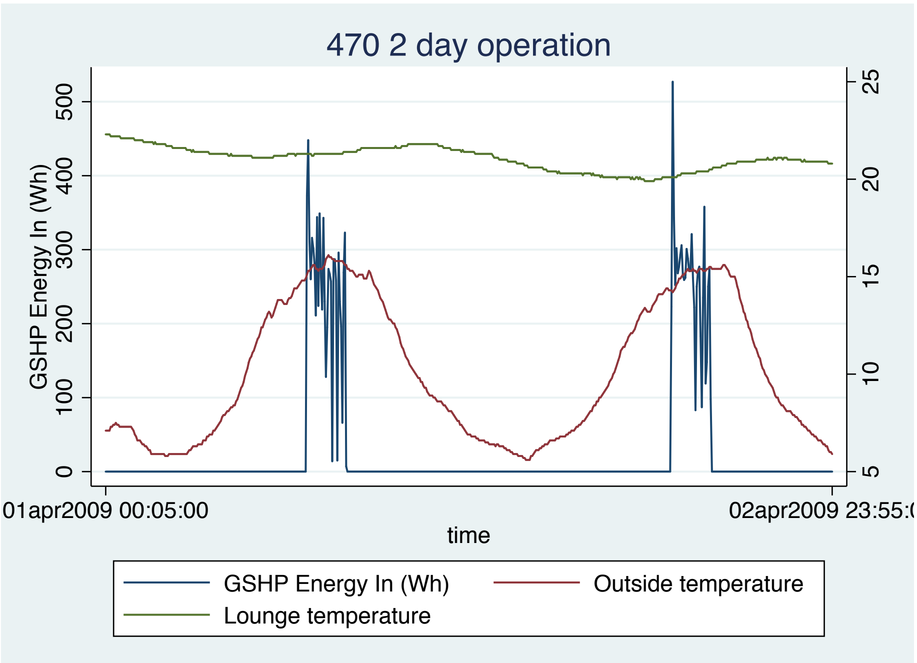


Figure 6—23 ID Code 470 - 2 day operation, heavyweight building

The heat pump runs once a day whilst retaining the lounge temperature at or above 20°C. The lounge temperature indicates high thermal mass capable of retaining heat for a 24 hour period. This long time constant reduces the chances of more rapid cycling and may be a contributing factor to the relatively high SPF_{H4} of 3.2.

We may compare Figure 6-23 with the operation of the ID Code 469 unit in a "detached house, converted in 1989 from timber frame barn", Figures 6-24 and 6-25.

Figure 6—24 ID Code 469 Lightweight building (EST Site Report, 2010)

The dwelling has a 12 kW heat pump for a total floor area of 44m² and is almost certainly oversized, has the same RDSAP band of D indicating a similar level of insulation to ID Code 470 yet results in cycling and an SPF_{H4} of 1.7 compared to 3.2, Figure 6-25.

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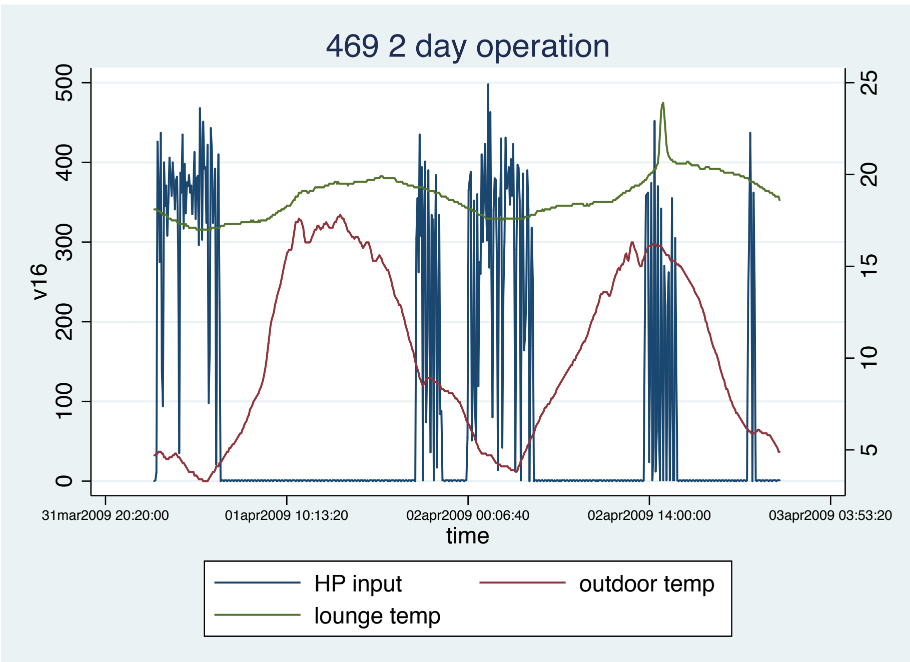

Figure 6—25 ID Code 469 - 2 day operation lightweight building, HP oversized

Other forms of heating such as gas boilers deal with oversizing through modulation, an option only available for heat pumps with variable speed compressors. There are only three such models in the trials, all air source heat pumps. It could therefore be concluded that, in the absence of variable speed compressors, heat pump over-sizing has had a significant negative impact on the overall trial results, a situation complicated by the inclusion of integrated backup resistance heaters for more than 50% of the trial heat pumps and where the control of backup switching is not transparent during operation. It would be easy to interpret this as an evidence-based warning against oversizing. But despite the lack of empirical evidence, there are good theoretical grounds for expecting heat pumps with variable speed compressors and to perform better when oversized than when closely sized since they can operate at a lower compressor speed. This illustrates the potential for drawing erroneous conclusions where empirical analysis is not complemented by a firm grasp of theory.

Designing the heat sink

UK heating design practice is historically associated with fossil fuel boilers and BS 5449:1990 flow and return temperatures set at 82 – 70°C, a 12K temperature rise across the boiler. From the 1st April 2005, Part L of the England and Wales building regulations required that all boilers be high efficiency condensing models where the return temperature needs to be below 57°C, the approximate dew point temperature of the flue gases. EN 677:1988 provided flow and return temperature guidance of 50/30°C for condensing boilers, the same temperatures as in BS EN 15502-2-1:2012, the current standard for condensing boilers.

BS EN 15316-4-2:2008, "Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 4-2: Space heating generation heat pump systems", offers no specific guidance on flow temperatures; the examples in the standard refer to sink temperatures ranging from 40 to 55°C, reflecting medium to high temperature emitters. Typically, a heat pump will generate a 5 to 6K temperature rise giving, under these circumstances, return temperatures ranging from 35 to 50°C.

The heat output of emitters is a function of the temperature difference between the mean emitter surface temperature and air temperature in the form of $Q = kA(t_m - t_a)^n$, where k is the heat transfer constant for the emitter, A the emitter surface area, t_m the mean emitter temperature, t_a the room temperature and n the index representing the convective component - the higher the value of n, the greater the convective output. Since UK manufacturers supply radiator output data based on BS EN 442-3: 2003 with a mean water to air temperature difference of 50K (based on a mean radiator temperature of 70°C) radiator catalogue outputs need to be corrected for all systems including those based on the traditional flow and return temperatures of 82/70°C.

When designing for sub-critical HFC refrigerant heat pumps, which have maximum flow temperatures of between 55 and 60°C, for a mean water temperature of 50°C and air temperature of 20°C (a 30K temperature difference) the catalogue output for UK standard panel radiators is reduced by about 50%, or, to achieve the catalogue output, the radiator needs to be double the catalogue size, (Myson, 2012).

The practical outcome is that when exchanging a boiler for a heat pump, existing radiators are far too small to provide adequate heat at low outdoor temperatures unless, perhaps, if originally oversized for intermittent operation. For all undersized radiators, whether existing or new, the occupant will either reset the heat pump to run at a higher temperature, usually maximum, or the heat pump controls will bring on any resistance backup; both courses of action resulting in lowering the SPF. Since the Carnot efficiency is a function of heat pump temperature rise, the lower the emitter temperature the greater the potential heat pump efficiency but the attendant need for larger radiator surface area or fan convector radiators, the so called "smart radiator" (Dimplex, 2010) but with additional parasitic losses from the convector fans.

Design guidance for underfloor heating, BS EN 1264-2:2008 + A1:2012, suggests a maximum floor surface temperature of 29°C, based on providing a maximum flow temperature of about 35°C, the ideal form of emitter for heat pumps. Underfloor heating is dominated by radiant heat output, requiring an adjustment to normal heating design calculations along with further adjustments for floor surface finishes. The resulting calculations are complex in comparison to those for radiator systems and, since the emitter pipes are buried in the floor, there is no option to replace an undersized system.

BS EN 1264-2 suggests the design and calculation methodology for the determination of floor thermal output be based on $Q = 8.92(t_{fm} - T_i)^{1.1}$ (W/m^2), where $(t_{fm} - T_i)$ is the average temperature difference between the heating surface and the standard indoor room temperature. The maximum output of an underfloor heating system at a 9K temperature difference (29 - 20°C) is about 100 W/m2 and is therefore limited to buildings with a reasonable standard of insulation and air tightness. Solid ground floors in particular need to be heavily insulated and upper timber floors require heat transfer plates inset with grooves for the piping to be fixed between the joists. Underfloor heating is therefore only an option for new build or when carrying out a full refurbishment or 'deep retrofit'. Because of these additional demands on the heating design calculations, specification and installation, it is generally provided by a specialist contractor and will entail the added complexity of managing yet another sub-contractor and supply chain.

Continuous or intermittent control

The EST trial literature presents no information on whether individual heating systems were set up for continuous or intermittent control. The confidential EST Technical Report of May, 2010 identifies five run-time operating conditions with 68% of installations in the continuous mode, Table 6-15.

Heat Patterns	Heat Pumps
Continuous	68%
Uni-modal	11%
Bi-modal	4%
Tri-modal	4%
Other	12%

Table 6—15 Operating conditions. (EST, Technical Report. 2010. p81)

We may attempt to analyse the impact of these modes of operation on seasonal performance. The confidential EST Trial Appendix 3 - Site data, provides information on electricity tariffs where there is a clear distinction between "economy" tariffs, E10 and E7, and single price tariffs. For ground source heat pumps, it is possible to identify the tariff structure for 37 heat pumps. The seasonal efficiency is tabulated against tariff in Table 6-16. As can be seen from the SPF means, there is no clear evidence that supports any one type of tariff. Even within these tariff types no clear picture emerges; the former example installations ID Codes 469 and 470, are both on Economy 10 but provide SPF_{H4} efficiencies of 1.7 and 3.2 respectively.

CODE	Tariff	SPFH4	SPFH5	SEEF
407	E10		2.1	2.1
417	E10		2.6	2.6
419	E10		3.0	3.0
420	E10		3.0	3.0
421	E10		1.7	1.7
430	E10	2.3		2.3
457	E10	2.5		2.5
469	E10	1.7	1.6	1.6
470	E10	3.2		3.2
	MEAN	**2.4**	**2.3**	**2.4**
CODE	Tariff	SPFH4	SPFH5	SEEF
414	E7		2.7	2.7
434	E7		2.0	2.0
451	E7	2.3	2.2	2.2
454	E7		2.5	2.5
459	E7		2.9	2.9
461	E7	2.5	2.1	2.1
466	E7		2.7	2.7
480	E7		3.0	3.0
	MEAN	**2.4**	**2.5**	**2.5**
CODE	Tariff	SPFH4	SPFH5	SEEF
411	S		2.3	2.3
412	S		1.9	1.9
413	S		2.3	2.3
415	S		2.4	2.4
416	S		2.3	2.3
431	S	2.8		2.8
435	S		2.2	2.2
438	S		2.1	2.1
439	S		3.4	3.4
452	S	3.3	3.0	3.0
453	S		3.3	3.3
455	S		2.4	2.4
456	S		2.2	2.2
481	S		2.4	2.4
482	S		1.8	1.8
	MEAN	**3.1**	**2.4**	**2.5**

Table 6—16 SPF comparison by electricity tariff

Economy 10 and Economy 7 tariffs are theoretically associated with a poorer SPF due to the need to supply heat at maximum output temperature during the limited run periods associated with the cheaper tariffs, however, since the SPF_{H4} outputs are so few, there is no evidence to support this theory. The responses to the confidential Open University questionnaires (OU, 2010 "Heat Pump user experiences, behaviour, perceptions and satisfaction") indicate that the majority of occupiers think they have controls set for continuous heating; the OU report does not attempt to verify this by relating, for example, heat pump operation to electricity tariff. The whole point of Economy 7 and 10 is to use the heat pump only when electricity is cheapest and therefore not continuously, particularly important where bivalent systems are installed. In addition to the five control categories, what the authors call "heat patterns", the internal temperature may be controlled by closed loop internal feedback only, in the form of a room thermostat, or by open loop weather compensation with some manufacturers providing additional feedback through a room thermostat. Again there is no explicit information on the control protocol for each system.

Kensa heat pumps (online), for example, provide the following online guidance:

> "In the UK, the use of banded tariffs such as Economy 7, or more especially Economy 10, makes the use of weather compensation on heat pumps unlikely to give any significant cost savings and can actually increase the cost of running a heat pump. This is because during a banded and cheaper rate of electricity it is desirable to run the heat pump at its maximum heat output so that as much heat is forced into the building as possible, whilst being careful not to overheat it in milder weather."

Since an off-peak tariff requires an oversized heat pump for intermittent operation, in principle, the choice between un-restricted and off-peak tariffs involves a trade-off between the capital and running costs of the heat pump although, as is evident, the EST trial provides little evidence to support this.

Weather compensation

For a fixed emitter size, the optimum control protocol is weather compensation. Emitters are designed for a heat output, based on mean water to room air temperature difference, to achieve the set-point room temperature at the design temperature difference. At all other times the mean water temperature is reduced to match the lower building heat loss thus increasing the operating efficiency of the heat pump. The commissioning of weather compensation control is by setting the controls to a pre-set compensation curve. These curves are plotted from calculations based on emitter heat output matched to building heat loss across the heating season for either a maximum mean or maximum return space heating temperature. Three such curves, based on examples with a 5K temperature drop and return temperatures of 50 (blue line), 40 (red line) and 30°C (green line), are shown in Figure 6-26, the enabling calculations provided in Table 6-17.

ts	te	tr	dt = (ts-te)	ts	te	tr	dt = (ts-te)	ts	te	tr	dt = (ts-te)
20	-10	50.0	30	20	-10	40.0	30	20	-10	30.0	30
20	-9	49.4	29	20	-9	39.6	29	20	-9	29.8	29
20	-8	48.7	28	20	-8	39.2	28	20	-8	29.6	28
20	-7	48.1	27	20	-7	38.7	27	20	-7	29.4	27
20	-6	47.4	26	20	-6	38.3	26	20	-6	29.2	26
20	-5	46.7	25	20	-5	37.9	25	20	-5	29.0	25
20	-4	46.0	24	20	-4	37.4	24	20	-4	28.8	24
20	-3	45.3	23	20	-3	37.0	23	20	-3	28.6	23
20	-2	44.6	22	20	-2	36.5	22	20	-2	28.4	22
20	-1	43.9	21	20	-1	36.0	21	20	-1	28.1	21
20	0	43.2	20	20	0	35.6	20	20	0	27.9	20
20	1	42.4	19	20	1	35.1	19	20	1	27.7	19
20	2	41.7	18	20	2	34.6	18	20	2	27.4	18
20	3	40.9	17	20	3	34.0	17	20	3	27.2	17
20	4	40.1	16	20	4	33.5	16	20	4	26.9	16
20	5	39.3	15	20	5	33.0	15	20	5	26.7	15
20	6	38.5	14	20	6	32.4	14	20	6	26.4	14
20	7	37.6	13	20	7	31.9	13	20	7	26.1	13
20	8	36.7	12	20	8	31.3	12	20	8	25.9	12
20	9	35.8	11	20	9	30.7	11	20	9	25.6	11
20	10	34.9	10	20	10	30.1	10	20	10	25.2	10
20	11	33.9	9	20	11	29.4	9	20	11	24.9	9
20	12	32.9	8	20	12	28.7	8	20	12	24.6	8
20	13	31.8	7	20	13	28.0	7	20	13	24.2	7
20	14	30.7	6	20	14	27.3	6	20	14	23.8	6
20	15	29.5	5	20	15	26.5	5	20	15	23.4	5

Table 6—17 Calculating return temperature settings against building temperature difference

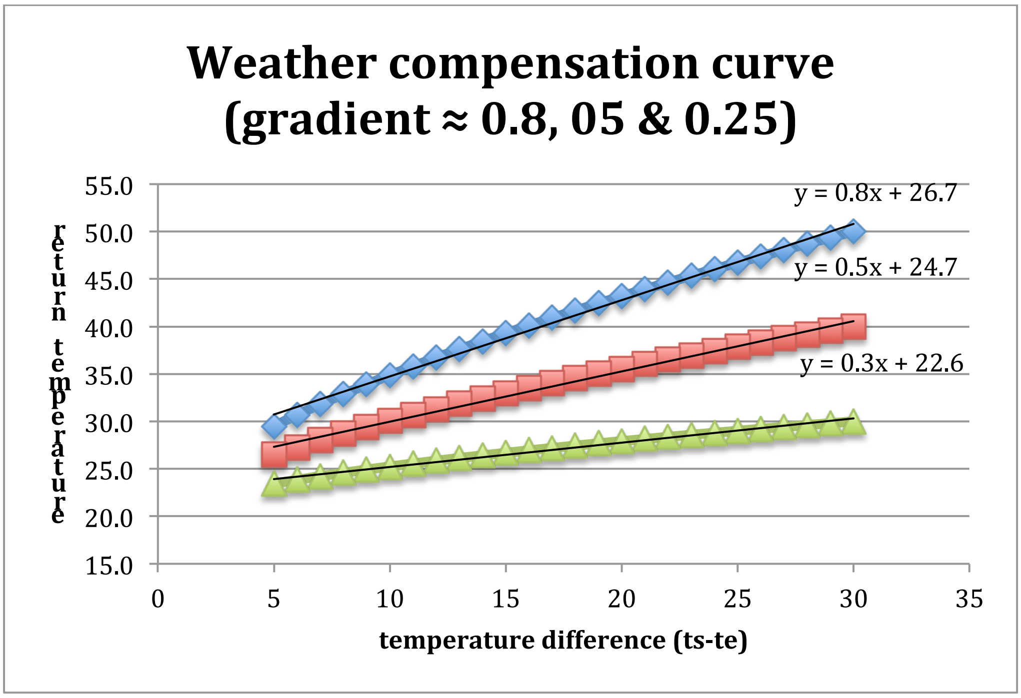

Figure 6—26 Weather compensation control curves: blue and red radiators, green underfloor heating

The gradient of the curve must match the heat loss conditions of the building and initial setup is usually based on the manufacturer's instructions, see for instance the Nibe Fighter 360P (Nibe, undated), Figure 6-27.

Setting using diagrams

FIGHTER 360P is equipped with outdoor temperature controlled automatic controls. This means the supply temperature is regulated in relation to the current out-door temperature.

The diagram is based on the dimensioned outdoor temperature in the area and the dimensioned supply temperature of the heating system. When these two values meet, the heating control's curve slope can be read. This is set under menu 2.1, Heating curve.

A suitable value is set using the knob on the front panel, Offset heating curve (38). A suitable value for floor heating is -1 and for radiator systems -2.

Offset heating curve -2

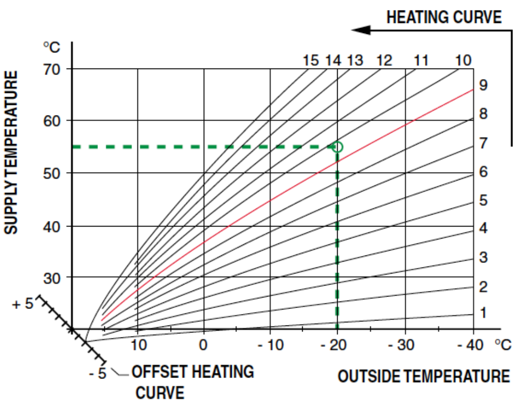

Figure 6—27 Nibe Fighter 360P Installation and maintenance instructions (p23)

Offsetting the heating curve from -2 to +2 will raise the design flow temperature from 55°C to 60°C. It must be evident that such an initial curve setting, carried out during commissioning, will be provisional in that it needs to be tested against outside weather conditions over winter. 15 bands are provided implying that fine-tuning will improve the seasonal performance; commissioning is therefore an on-going process. It is at this point that the occupant will most likely be responsible for fine-tuning and have to make sense of operating instructions that must appear almost incomprehensible to the average occupant. Changing the heat curve, or "offsetting", is shown in Figure 6-28, where the reference to "menu 2.0" is part of a 5 main menu flow diagram with 55 "submenus" (Nibe, undated, pp24-25).

E **Offset heating curve**

This knob is used to change the room temperature. Turning clockwise in-creases the room temperature. When the knob is turned menu 2.0 is shown on the display screen and the value for the calculated supply temperature changes.

Figure 6—28 Nibe Fighter 360P Installation and maintenance instructions (p5)

These Nibe Fighter 360P installation and maintenance instructions have 10 pages (pp23-33) explaining control setup. The installer will, it can be imagined, follow a checklist derived from manufacturer's training or just switch on and hope! To presume that the occupant, without any technical knowledge, could adjust the controls in order to fine-tune the system seems somewhat optimistic. What is needed to resolve this conflict is adaptive, self organising controls.

Another manufacturer, Heat King (online), provide the following example:

"Weather compensation: for units preset to operate with a radiator based wet system, weather compensation is programmed into the controller. This facility is not available with underfloor systems as the minimum coolant operating temperature is 35°C, preset as standard for underfloor systems. Weather compensation should not be activated when the heat pump is used for Domestic Hot Water unless a Variable (or fixed) set point kit is fitted. The Variable set point kit allows a variable second temperature set point to be programmed and used for a night time setback function (Space Heating) or for DHW, e.g. under floor Heating at 35°C, DHW set point at 50°C. *Both set points can be altered by the installer to suit their particular installation* [author's italics]. The design of this function is such that the primary set point can be used with weather compensation enabled (Programmed by user), but the second set point has the weather compensation disabled. This is useful for DHW when the set point should not be affected by ambient conditions," p14.

Such a description is badly in need of 'Plain English' and manufacturers in general could benefit from the language rules developed for maintenance manuals by the Aerospace and Defence Industries of Europe (ASD) known as "Simplified Technical English", (ASD, 2013)

The Worcester Bosch Greensource user manual (Worcester Bosch) describes weather compensation as the default control allied to an internal sensor for feedback:

"Weather-compensated control with room temperature dependence means that a temperature sensor (accessory) is located in the lead room of the building. The room temperature sensor is connected to the heat pump and signals the current room temperature to the control unit. The room temperature sensor influences the heating curve flow temperature. The flow temperature is reduced if the actual room temperature is higher than the selected temperature. A room temperature sensor is appropriate if, apart from the outside temperature, other factors influence the temperature inside the building, e.g. open fireplace, fan heater, building subject to wind influence or direct sunlight," p9.

In contrast, the Calorex heat pump is designed for connection to standard control solutions based on internal air temperature control via a room thermostat. The Installation/technical manual describes "economy/high" temperature control which is left for the installer/occupier to adjust in order to reduce system temperatures to the minimum for comfort, (Calorex).

It would appear that control setting is wholly dependent on the individual manufacturer's design and for those fitting a variety of heat pumps, requires an understanding of these various forms of control in order to commission. For those more used to boiler commissioning, weather compensation is still a specialist control option in the UK, in an industry dominated by a '*two or three-port valve with room and cylinder thermostat*' standard, commonly known as the S and Y plan, (Honeywell Sundial, online).

The issue of installer training is paramount. No generic heat pump training will provide the specific knowledge required for commissioning all heat pumps. For the occupant in particular, it appears that control setting and fine-tuning could lead to entirely inappropriate functioning.

Control and the building time constant

None of the manufacturers appear to have considered the relationship between control and thermal mass. We have seen, in Table 6-14 and Figure 6-21, that long time constants are associated with both super-insulated lightweight and heavyweight envelopes. Weather compensation will call for changes in mean emitter temperature in response to changes in outdoor temperature. For predominantly maritime climate zones, with their relatively rapid changes in winter temperature, there is the issue of time lag between the change signal and the impact on internal temperature. For very long time constants combined with internal heat gains, continuous heating by weather compensation is likely to lead to temperatures exceeding the set point. The same should apply to night-time set back, a proposition confirmed by Boait (Boait, et al, 2011).

Domestic hot water

The trial provides three types of hot water storage utilised for domestic hot water: within the heat pump unit as a built in calorifier, as a thermal store with an embedded cylinder or the traditional separate hot water cylinder. Whilst some systems have no hot water boost (an immersion heater), where it does occur, data collection on resistance boosting varies between integrated systems with no separate meter, separately metered boost to the entire system or a metered cylinder immersion heater.

UK Health and Safety Executive (HSE) guidance states that, for hot water storage, "Outgoing water should be at least 60°C." (HSE, online). In contrast, EST field trials of residential hot water cylinders provided a mean temperature of 52.9°C, well within the output range of HFC refrigerant heat pumps (EST/DEFRA, 2008). However, where space heating is at a lower temperature or weather compensated, this will require a dual or change-over temperature control function which, due to higher DHW temperatures, will impact on SPF. Thus, in theory, combined space heating and domestic hot water systems should have a lower SPF than space heating alone but as we have seen, there is little correlation shown, due no doubt to the trial mean domestic hot water temperature which was measured at 46°C (EST, Appendix 3 - Site data, 2010, confidential).

Whilst domestic hot water is would normally be achieved at the maximum heat pump output temperature and therefore minimum Carnot efficiency, hot water generation may also lead to rapid cycling as observed in Boait's hot water production study (Boait, et al, 2012):

> "The large number of heating cycles performed by the heat pumps due to the low value for water temperature set point hysteresis of 4°C [the range may extend to 6°C depending on the manufacturer] employed by the control system. For comparison, the bimetallic strip thermostats used in the gas fuelled systems with simple control had a hysteresis of 10°C. The effect of this was that the heat pumps performed 4 or 5 heating cycles per day even at low levels of usage, rising to 7 - 10 per day for usage above 100 litres."

The paper goes on to describe the losses associated with cycling.

Occupant perception

The confidential Open University report for the EST (OU, 2010) covers occupant perception of their understanding of controls, although the results are somewhat contradictory:

> "Overall…. 64% of all heat pump users have at least a fair knowledge and understanding of their heat pump system. However, there is a strong contrast in knowledge and understanding between the social housing residents, where 60% claim that they have little or no knowledge or understanding of their heat pump system, and private householders, where 80% claim that they have a lot or fair knowledge or understanding of their heat pump system (only one social housing resident claimed to have a lot of knowledge/understanding)," p14.

It is not surprising that private householders, who after all have made the decision to install a heat pump, should claim to "have a lot or fair knowledge or understanding".

Contrast this statement to:

> "One key finding from this survey work is that over two fifths of users (44%) reported

uncertainties about using controls to operate the heat pump system most efficiently and nearly a third (30%) had difficulties in understanding instructions on operating and using the system," p16.

Whilst 55% of all respondents, "Changed/reprogrammed the (user) controls on the heat pump unit", there is no evidence that the changes that they made were to the good. Indeed, the author's analysis of the data from individual installations shows that a number of the systems have control faults such as:

- Heat pump runs continually even in August and where ticking over accounts for 16% of the input energy on top of a 20 minute run every 3 hours when there is no obvious load (416 - owner occupier)
- space heating on continuously all year even when outside temperature is in the high 20°Cs (417 – owner occupier)
- radiator based space heating operating in August although flow temperature does not exceed 25°C (422 – owner occupier)
- immersion heater operating every 40 minutes (487 – owner occupier)
- space heating turns on twice a day between 04.00 to 05.00 and 15.40 to 16.30 even during the summer (463 - social tenant).

Whilst this is not an exhaustive list and just a few examples, it does raise concerns about the uncritical acceptance of occupants' claims to knowledge of controls.

Secondary heating

Secondary space heating remains common in older buildings in the UK, providing occupants with a living room-only alternative heat source or the option to support the central heating system. 81% of the dwellings with ground source heat pumps have some form of secondary heating whether electric, gas or open fire, in comparison with 54% of air source. The secondary heating provides, potentially, ambiguity in the allocation of living room temperature and conflict in the control hierarchy where space heating control is by room thermostat. For heat pumps with resistance heaters and secondary space heating we have, effectively, trivalent installations. For those in fuel poverty, the living room "fire" is an alternative to whole house central heating and may conflict with the role of "trial test house", a situation borne out in the author's analysis of run times; some dwellings barely use the heat pump for space heating. The Open University reports that 31% of households used secondary heating.

Multiple contracts

Heat pump installation tends to complexity in comparison to gas boiler installation. In a plumbing industry that has trained and then feasted on the simplicity of combi boiler installation, "knock a hole in the wall for the flue and connect the pipes", the ground source heat pump introduces multiple layers of complexity in design, installation and controls. With the combi, the boiler industry has successfully integrated pretty much all of the components associated with heat generation and circulation for gas central heating into a single box, with a handful of connections. There are fundamental reasons for expecting such integration to be much more difficult to achieve with heat pumps.

Ground source heat pumps may require a ground loop or borehole contractor, an underfloor heating contractor, a heat pump contractor capable of matching building heat loss with heat pump selection, ground loop sizing, heating system design, control specification and commissioning. For feedback on system operation, a monitoring protocol must be included, especially with regard to any payments under the UK Renewable Heat Incentive (EST RHI, online). Finally the heat pump must be commissioned and the handover processes include some advice to the client on operation and control settings. These complex inputs will impact directly on seasonal performance.

Summary

The EST trial consists of a heterogeneous heat pump selection with a mixture of source and sink designs, backup heaters, control systems and monitoring protocols. Its primary efficiency metric, seasonal performance (SEFF) tells us little about the relationship between design, installation and performance. The taxonomic analysis reveals two fundamental system types, monovalent and bivalent but when looking for patterns based on thermodynamic principles we are consistently struck by the lack of a heat pump-only output across all systems whether at SPF_{H2} or just heat pump and backup at SPF_{H3} or heat pump, any backup and circulation pump at SPF_{H4}. All systems have been analysed by EST/DECC at SEFF irrespective of auxiliary heating, whether with an all-system backup or a mono-energetic domestic hot water immersion boost, and further complicated by domestic hot water cylinder losses. It would appear that consistent monitoring is the primary issue for trial design in understanding heat pump performance. Even at SPF_{H3} or SPF_{H4}, there is the need to analyse the reliance on backup since undersized or poorly controlled bivalent systems will underperform irrespective of how well the actual heat pump is working. To minimally understand just the heat pump, what is needed is SPF_{H2}, the heat pump only with its source fan or pump. However, the heat pump is part of a system, all the components of which need to work effectively together to achieve the objective – cheap, low carbon heat. This suggests that designers will always need SPFs calculated on multiple system boundaries to understand what is going on.

In addition to inadequate monitoring, we have identified a range of issues which impact on efficiency including:

- the mismatch between heating load and heat pump output
- the potential mis-reporting of slinky ground loop length and the question of correlation between ground loop length and heat pump output
- the potential for a negative impact from buffer vessels
- the role of thermal mass and its impact on both the addition of a separate domestic hot water load and on optimising control for thermal comfort and energy efficiency
- the need for clarity in manufacturers' instructions for commissioning controls and the need to rethink controls from the occupants' viewpoint.
- For ground source heat pumps in particular, these issues could be linked to the additional contractual complexity associated with designing and managing groundworks, whether ground loops or boreholes.

For all heat pumps, when considering the selection of the heat pump output, design of installation, choosing a controls protocol and commissioning the entire system, we are looking at a range of vocational education and training (VET) issues which must be addressed if heat pump performance is to improve. This, it would appear, was the main finding of the UK trials for DECC. The UK trials have led to the establishment of a system of registration and training under the auspices of the Microgeneration Certification Scheme (MCS, online). Within the European Union a similar process has been established under the EUCERT-HP, the European heat pump installer certification scheme (EHPA, online). The next chapter will analyse the contents of the MCS scheme in terms of pre-requisites for access to training and, in particular, the knowledge, skills and competences associated with heat pump installation design.

Chapter 7 EST, MCS and VET

Introduction

The EST report on the UK trials, (EST, 2010) provides the following series of quotes which emphasise many of the failures in design practice identified in Chapter 6 and the need to work with all stakeholders to improve heat pump training:

> "Heat pumps are sensitive to design and commissioning. The field trial covered a variety of early installations, many of which failed to correctly apply the heat pump. This result emphasises the need for improved training," p6
>
> "Among the 83 sites we monitored, there were good, average, and poor performing sites. This variation in performance has been influenced by a number of factors, including system design (sizing of the pump, and type and size of heat source and heat sink), system installation, and customer behaviour," p18.
>
> "Heating controls for heat pump installations have to be comprehensively reviewed. There has been a failure to explain proper control requirements to both installers and heat pump customers," p7.
>
> "Responsibility for the installation should be with one company, and ideally be contractually guaranteed to ensure consistency in after-sales service," p7.
>
> "We are actively working with the MCS and the heat pump industry to improve the availability of training for heat pump installers, including skills-based training courses," p22.

The aim of this chapter is to review MCS documentation, apply it to the EST trial analysis in the previous chapter and to comment on the vocational education and training (VET) implications.

Where reference is made to online resources subject to updating, such as webcasts, product or cost data from the internet, footnotes are supplied with the web address URL and date of access.

Microgeneration scheme

The DECC detailed analysis (Dunbabbin & Wickins, 2012, p2) refers to the training development work undertaken with the Microgeneration Certification Scheme (MCS) in the time since the initial 2010 trial analysis:

> "This site-by-site analysis has formed the basis of extensive discussions with the heat pump industry. As a result, the Microgeneration Certification Scheme (MCS) has drawn up new standards for the installation of heat pumps with <45kW heating capacity. These new MCS standards were launched in September 2011."

The MCS is an installer registration scheme, which requires quality assurance and appropriate training evidence from applicants for individual renewable technologies. Each renewable technology has a working group whose objectives and scope are, *inter alia*:

- "To develop and maintain Product and Installer standards for the technology under consideration
- To conduct a technical review of issues with, and proposed changes to, those Standards
- To ensure the Standards are up to date and fit for purpose" (MCS Working Group Terms of Reference, 10/01/2011, p2)

The scheme is built around product manufacturers and existing installers and the knowledge component is aimed primarily at the "*Experienced Workers Route*", that is, those without formal design education but with experience in central heating design and installation (MCS Heat Pump Working Group minutes, 13/12/2012, p7).

The MCS heat pump design guide proper is "Microgeneration Installation Standard: MIS 3005". The description that follows is based on Issue 3.2 (MCS/DECC, 2013). The installation standard is supported by guidance notes, ground loop sizing tables, an emitter guide and heat loss calculation software. These standards and guides are freely available online and subject to continuous update; at the time of writing, having last been updated on 22/07/2013. Knowledge standards are under continuous review managed by the MCS Heat Pump Working Group. To support the standards, MCS have developed "*reference materials*" in the form of webcasts, online software and open-source spreadsheets.

The MCS "Installer certification scheme requirements, Issue 3.2" outline the following with regard to training, (MCS, 2013, p22):

> "All staff employed in installation, set to work and/or commissioning activities must have received adequate training in each of the areas/operations in which they are involved.
> The Company must have a training record for each employee which details training received, and any qualifications or certificates held by the individual. The record should be signed or verified by the employee.
> The Company must have a record detailing the MCS related activities for which each individual is approved on the basis of their competence. The competencies required for installers are detailed in the relevant installation standards."

Domestic heating work is generally not the province of professional building services engineers, in particular, chartered engineers who would have studied the CIBSE guides as undergraduates. Whilst larger companies may employ heating designers, the demographics of the UK construction industry show that:

> "A feature of the sector is that there are a small number of large firms and a very long tail of small firms. Across the construction sector as a whole there are approximately 365,535 enterprises. However, the vast majority of companies in the sector are small, with over 93% employing less than 10 employees."
> (ConstructionSkills, 2010. p18)

A similar picture emerges from SummitSkills, the building services Sector Skills Council, where, for example, the Scottish Sector Profile 2011 (Alliance of Sector Skills Councils, Scotland, 2011, p4) states:

> "The Building Services Engineering sector has a relatively high proportion of very small workplaces, those with one to four employees, with 71% of workplaces fitting this category, compared with 59% of all Scottish workplaces."

Small companies will tend to be dominated by craft and advanced craft trained occupations (NVQ Levels 2 and 3), rather than by Level 4 Higher National Certificate/Diploma (Incorporated Engineer) or Level 7, the Chartered Engineer (C Eng or Eur Eng). An internet review of MCS heat pump short course training providers indicates that the minimum prerequisite is NVQ Level 2, although many will accept those with no formal qualification but with appropriate experience in recognition of the "experienced worker route". The BPEC plumbing NVQ Level 2, available online from

Ofqual (Ofqual[a], 2012), has three units directly related to hot water and central heating, however, their focus is on "understand" the technologies and "install and maintain"; there is no mention of engineering design for assessing loads and sizing components. Much design work will be based on 'rule of thumb' derived from custom and practice.

The MCS webcasts are aimed at designers and installers who wish to register with the scheme. They were developed from a series of DECC-funded training workshops for designers and installers and presented by David Matthews, CEO of the UK Ground Source Heat Pump Association (GSHPA, online). Both the technical content and the commentary indicate that the audience would be expected to struggle with, for example, the difference between kW and kWh, with heat loss calculations and annual energy costs. To replicate the webcast content onto the MCS supplied spreadsheets also requires experience of computers, spreadsheet layout and functions not necessarily part of the skill-set of "experienced workers" or covered by NVQ "functional skills" (OFQUAL, 2010), formerly "key skills" requirements. Perhaps in recognition of this, MCS have recently introduced an online heat loss and energy assessment calculator (MCS, heat pump software, online).

The aim of this chapter is to review these "standards for installation" and "reference materials" to identify the engineering knowledge requirements for those designing, installing and commissioning heat pumps. Specifically, the objectives are the analysis of MIS 3005 in terms of the potential designer, what must be understood in order to comply with the specific requirements for:

- Building heat loss, domestic hot water demand and the resulting annual energy assessment calculations
- Matching these heat losses to manufacturers' data for heat pump selection
- Ground loop design procedure and the associated commissioning demands
- Control specification
- Hand over procedure

The results will be cross-referenced to the outcomes of Chapter 6, the analysis of the EST trial data.

Much of the chapter follows the calculation method set out in the MCS documentation, however, it is the author's view that such calculations and importantly, their iterations which result in the synthesis of design options, are outside the educational scope of the untrained "experienced worker" as well as the NVQ Level 2 operative and are unlikely to be developed during the current short provision of just a few days. They indicate the need for critical review of designer qualifications.

Heat loss calculations and annual energy

MIS 3005 requires that heat loss calculations "should be performed using a method that complies with BS EN 12831".

BS EN 12831: 2003, "Heating systems in buildings - Method for calculation of the design heat load", is clearly written with the professional engineer in mind; "*Table 1*", for example, lists 28 "*Symbols and Units*" supported by "*Table 2*" with its 30 different "*Indices*". BS EN 12831 is a room-by-room method based on internal and external temperatures, rather than a whole building approach, and refers to "*National Standards"* for these reference temperatures. However, the supporting webcasts make it clear that the design procedure is based on existing housing and not those built to current low energy standards. MIS 3005 is essentially aimed at the boiler replacement market where the space heating load is assumed to far outweigh that of domestic hot water.

MIS 3005 provides individual room temperatures and outside design temperatures for different UK locations and clause 4.2.1 c) states that:

> "A heat pump shall be selected that will provide at least 100% of the calculated design space heating power requirement at the selected internal and external temperatures, the selection being made after taking into consideration the space heating flow temperature assumed in the heat emitter circuit and any variation in heat pump performance that may result. Performance data from both the heat pump manufacturer and the emitter system designer should be provided to support the heat pump selection," p13.

The heat pump alone must be able to supply 100% of the load even where there is a built in resistance heater. It is therefore apparent that the designer must understand the all factors which impact on heat pump power selection associated with fabric heat losses, u values, thermal bridges, individual room losses and gains driven by temperature differences across partition and external walls, and room ventilation rates. The designer must be able to interpret individual manufacturer's performance data and correlate emitter selection with likely SPF in order to support the heat pump selection.

In apparent contradiction, MIS 3005 then states in *Notes on section 4.2.1.* (p14):

> "Sizing a system to precisely 100% as defined in section 4.2.1 part (c) will require supplementary space heating for the coldest 1% of the hours in a year. In addition, the system may require the use of supplementary heating if:
>
> - The building is being heated from a cold state;
> - The desired heating mode is not continuous, such as bi-modal heating or heating using a split-rate tariff;
> - Large quantities of domestic hot water are required frequently during cold weather.
>
> Installers trying to design a system capable of achieving these requirements without supplementary heat should consider increasing the heating capacity of the heat pump. The clause in section 4.2.1 (c) requires the CIBSE external design temperature to be the temperature at which the heat pump heating capacity at least matches the building design load."

There are a number of issues raised by this statement. There is an underlying assumption that the design default is continuous heating, tenuous at best given the UK preference for intermittent heating, combined with the proliferation of Economy 7 and 10 tariffs specifically aimed at reducing the impact of electrical loads during peak demand times. All dwellings will, at some point, require heating from cold during cold weather, and all buildings will, again at some point, be subject to the 1% below design temperatures. Since all systems fall into these categories, all systems will require backup or significant oversizing. We will return to the issue of domestic hot water but it is clear that these heat loss and plant sizing issues can only be addressed if the designer has the requisite knowledge and the ability to make informed decisions.

The cost of power - heat pumps versus boilers

The sizing heat pumps, their power output, has a far more signicant impact on system cost than that of gas boilers. Heat pump cost per kilowatt is directly related to the additional sizing requirements for their component parts and, in the case of ground source units, the length of ground loop. One would expect tender documents to reflect this since cost alone remains significant in tender selection. For domestic gas boilers the only difference between boilers of different output is in the heat exchanger size. For combi boilers, however, since they are designed for maximum output during DHW production (often between 25 and 30 kW) the only change for space heating is in the electronic control of gas rate.

Some understanding of the difference in cost between heat pumps and gas boilers, based on cost per kW output, is presented in Figure 7-1 where the vertical axis represents power output and the horizontal axis unit cost. Air source[12], combi[13] and system boiler[14] prices are from current UK sales websites where gas boilers, in comparison to heat pumps, show a minor price rise as power increases. The costs for ground source heat pumps are based on historical American data (Kavanaugh, et al, 1995) where neither currency nor cost have been normalised from dollars to 2013 exchange rate in GB pounds but it is clear from the graphic that ground source units exhibit similar gradient features to air source units. Contractors are therefore under pressure to undersize heat pumps for economic reasons that do not apply to gas boilers.

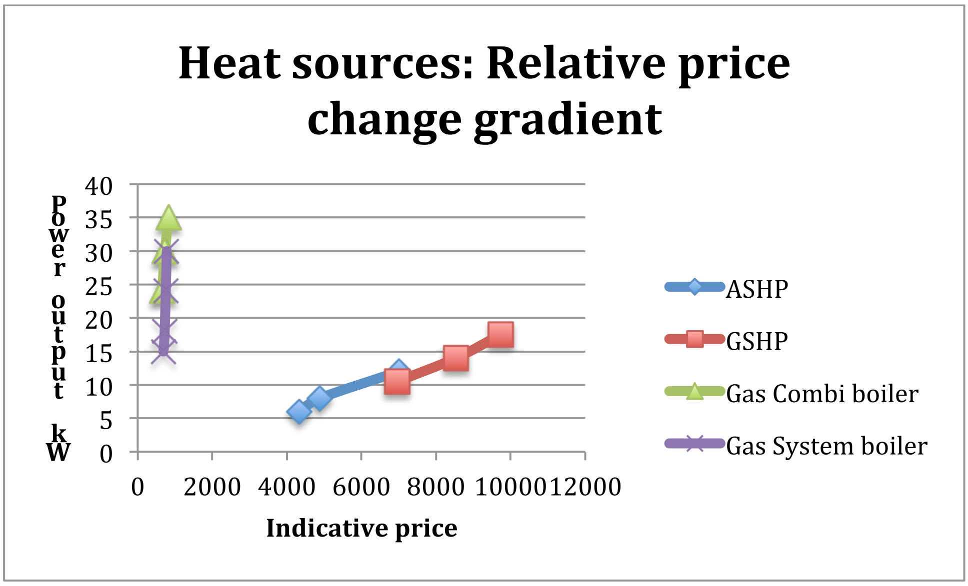

Figure 7—1 Price comparison between heat pumps and gas boilers

Calculating heating load

Heat loss calculations are supported by the MCS reference materials that include a series of webcasts. "*Presentation 3, BS EN 12831 and sizing*"[15], provides a description of the heat loss design process and refers to three supportive documents: BS EN 12831, the CIBSE Guide A and the CIBSE Domestic Heating Design Guide (DHDG) (CIBSE 2012). Given the engineering knowledge required to understand and interpret BS EN 12831

[12] http://www.plumbcenter.co.uk/en/renewables/air-source-heat-pumps/ [accessed 23 September 2013]
[13] http://www.discountedheating.co.uk/shop/acatalog/Ideal_Logic_Plus_System_Boilers.html [accessed 23 September 2013]
[14] http://www.discountedheating.co.uk/shop/acatalog/Ideal_Condensing_System_Boilers.html [accessed 23 September 2013]
[15] http://www.screencast.com/t/4BZqepKX [accessed 23 September 2013]

and the CIBSE Guide A, it is most likely that those seeking support with heat loss calculations, will focus on the DHDG.

With regard, for example, to pre-heat and boost requirements, CIBSE Guide A, Environmental design (CIBSE, 2006), dedicates an entire chapter of 96 pages to "*thermal response and plant sizing"* based on: type of thermal input, surface finishes, thermal properties of the construction, thickness of the construction and furnishings within the space, in order to assess whether the building is slow or fast response, heavy or lightweight. Engineers interpret these factors in order to reach a decision; a choice is made over whether the impact of emitter convection or radiation is dominant (F_{1cu} and F_{2cu}), the room admittance factors (effectively the thermal mass effects) allied to the u values and ventilation conductance to calculate the thermal response factor (f_r), defined as $f_r = \frac{\Sigma(AY)+C}{\Sigma(AU)+C}$, in order to calculate the plant sizing ratio F_3.

A rather more user-friendly, although still highly technical, description of this CIBSE design process is provided by Moss (2003, pp14-29) in his textbook aimed at building services students on HNC/D and degree courses. In contrast to this engineering approach, Worksheet Five, p53, in the DHDG simply states an allowance of 15% for intermittent heating and 10% for distribution losses.

MCS have provided online software[16] to simplify heat loss calculations by removing the need to manipulate spreadsheets. It is an MCS requirement that both the heat pump power (kW) and the annual running costs (kWh/yr x unit marginal cost) are presented to the client and to achieve the latter, the software requires the 'degree day' location. Degree day annual energy calculations are based on the relationship between diurnal and design base temperatures, plant kiloWatt output, plant efficiency and any corrections for intermittent operation to achieve annual kiloWatthours. The room by room calculations require areas, their U values, the design temperature difference and whether the property is in an exposed location (add 10% to the calculated heat loss), has underfloor heating, (no impact), and intermittent heating (add 12 - 15%). In terms of the "experienced worker", that is someone with little or no formal training, we are in the zone of 'gigo', garbage in, garbage out, an observation and perhaps a warning made by David Matthews in his MCS webcasts.

[16] http://www.microgenerationcertification.org/mcs-standards/installer-standards/mcs-heat-pump-software [accessed 23 September 2013]

A fully worked example is probably the best method for those struggling with such calculations. The main reference within the webcasts is to the CIBSE DHDG, which unfortunately, neither provides a dimensioned building nor a fully worked whole house room-by-room example. The webcasts that discuss the heat loss spreadsheets, obviously developed before the online calculator, also fail to provide a room-by-room example, instead relying on a whole house example based on down stairs and upstairs. Additionally, the webcast spreadsheets are also different from those supplied for download.

With regard to MCS short-course providers, some have either: “no specific entry requirements for attending”[17] or require that: “Operatives should have a good working knowledge of heating/hot water systems and design with an NVQ Level 2 qualification *or equivalent experience*” [18] (author’s italics). It should be noted that the former is for a 3 day course and the latter for 4 days. It is telling that the second short-course provider appends a warning to their website regarding calculations:

> “Calculating building load and heating system design is not covered in any great depth in this course. Operatives without this knowledge should employ the services of a suitably qualified person, seek manufacturer’s guidance, or take the Logic4training Heating and Hot Water Systems and Safety course.”

The design output of the heat pump is based on the system flow temperature and either the ambient design temperature for air source heat pumps, or a ground loop temperature of 0°C for ground source. This will require the designer to interpolate across a manufacturer’s data tables or graphs to identify the output at these design conditions. If, for example, the building heat loss is 10kW at (-3.9)°C in Glasgow for an air source unit, then the heat pump selection must be interpolated from 10kW at between (-2) and (-7)°C, the nearest EN 14511 COP test temperatures. We may find that the heat pump is no longer sufficiently large since most manufacturers will publish nominal outputs at 7°C, such as the Ecodan Monoblock[19] advertised at 5 kW output. Its output at (-4)°C is not obvious since there is no published test data below (-3)°C.

[17] Easy MCS Academy: http://www.easymcsacademy.com/renewable-courses/heat-pumps-2.html [accessed 23 September 2013]
[18] Logic 4 Training: http://www.logic4training.co.uk/renewables-courses/heat-pumps-courses/Heat-Pump-Installation-training-course?gclid=CIbQ8riq07gCFQSS3goduTMAjw [accessed 23 September 2013]
[19] http://domesticheating.mitsubishielectric.co.uk/files/library/files/ecodan/Brochure%20downloads/Ecodan%20PUHZ-%28H%29W50-140VHA2-YHA2%20%28FTC4%29%20-%20JAN%2013%20LoRes.pdf [accessed 23 September 2013]

Having selected a heat pump the installer has to calculate the annual running cost. This will require the use of the "Heat Emitter Guide for Heat Pumps"[20]. The guide provides a "*likely SPF*" based on EN 14825 (latest version 2012), part load efficiency or SCOP, where the heat pump is sized for 100% of the load with national weather data for Leeds, UK. In addition to supplying the SPF for annual energy use, the guide provides sizing information for different emitters, based on the building heat loss coefficient (W/K) and heating circuit flow temperature, including various types of radiators, fan coils and underfloor heating.

MIS 3005, section 4.2.1.(c) makes no mention of the additional domestic hot water load although the standard implies that hot water is integrated and section 4.3.1 requires the designer to provide:

> "An estimate of annual energy performance shall be calculated or obtained and shall be communicated in writing to the client at or before the point at which the contract is awarded. Separate calculations for space heating and for hot water shall be performed and subsequently added together to give a combined annual energy performance figure," p25.

MIS 3005 clause 4.2.15. a) demands of the assessment of annual energy:

> "The total heating energy consumption over a year (in kWh) for space heating and domestic hot water shall be estimated using a suitable method. The calculation shall include *appropriate consideration* [author's italics] of internal heat gains, heat gains from solar insolation, local external air temperature and the heating pattern used in the building (e.g. continuous, bi-modal, with an Economy10 tariff or otherwise)," p20.

How a designer is to evidence "*appropriate consideration*" for a range of occupant-driven energy flows is not discussed. The webcast series does not include any consideration of internal or solar gains and the heat pump is modelled for continuous use.

Domestic Hot Water

Possibly the most difficult issue to decide on, without simply resorting to rule of thumb, is that of sizing for domestic hot water. MIS 3005 demands that hot water design:

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[20] http://www.microgenerationcertification.org/images/MIS_3005_Supplementary_Information_2_-_Heat_Emitter_Guide_v2.0_Print_Version.pdf [accessed 23 September 2013]

> "should be based on an *accurate assessment* of the number and types of points of use and anticipated consumption…. Making *appropriate adjustments* for the intended domestic hot water storage temperature and domestic hot water cylinder *recovery rate*," [author's italics] p17.

Such "assessments" and "adjustments" are surely beyond the skills set of the unqualified "experienced worker".

The implication of MCS 3005 is that space heating load is dominant and that the control system will provide for one load at a time, switching between space heating and domestic hot. Whilst there is no addition of a 2 to 3 kW power load for DHW based on the size of the dwelling (DHDG, 2011 p55), the DHW energy demand is combined with the space heating energy demand for inclusion in the total annual energy load (kWh).

The standard refers to three documents for "additional information":

> "BS 6700: Specification for design, installation, testing and maintenance of services supplying water for domestic use within buildings and their curtilages; EN 806: Specifications for installations inside buildings conveying water for human consumption; and studies conducted by the Energy Saving Trust and Department of Energy and Climate Change [DEFRA], for example Measurement of domestic hot water consumption in dwellings (Energy Monitoring Company) March 2008," p17.

BS EN 806 is an overall system design and maintenance series rather than a sizing guide to flow rates and storage and does not provide a litres per person assessment. BS 6700:2006+A1:2009 (amended in 2009 and superseded, withdrawn, replaced by the BS EN 806 series) suggests 35 to 45 litres/person per day as the hot water requirement. The EST/DEFRA report provides actual hot water use across a sample of 112 dwellings where the mean consumption was 122 ± 18 litres/day, along with a mean hot water temperature for 69 regular boilers of 52.9 ± 1.5 °C. Where there are less than 5 occupants, the report suggests 40 + 28N as the daily consumption in litres, where "N" is the number of occupants. For a two bedroom, 3 person house, this provides 124 litres of hot water per day (EST/DEFRA, 2008).

The MCS webcasts support BS 6700 at 45 litres/person, whilst suggesting designing for the property, not the current number of occupants, and thus the number of bedrooms.

Hence: “number of bedrooms +1 = occupants”; for a two bedroom house this provides 135 litres.

Where the heat pump cannot supply hot water at 60°C, the Spreadsheet 5 webcast[21] provides the additional calculations for daily immersion boost and annual energy demand. Apart from the references to specific heat capacity and conversion from Joules to kWh, the designer must also understand that the hot water cylinder needs to be supplied at the maximum heat pump flow temperature which ranges between manufacturers from about 50 to 55°C. Since this temperature may be above that for the emitters, the heat pump power output must be re-interpolated for DHW from manufacturers’ efficiency data. This power difference becomes most evident where heat pump flow to the hot water cylinder is at 50 or 55°C and space heating at 35°C and will entail an adjustment of manufacturers’ output data and SPF based on the MCS Heat Emitter Guide. This process of assessing heat pump efficiency under two separate temperature flow conditions is unique to heat pumps.

The daily immersion load calculations are based on a temperature rise from 45°C (a 5K temperature difference between the primary flow at 50°C and secondary water) to 60°C, the minimum to control legionella. Whilst the calculation method is correct, the immersion heater will require a time control otherwise it will operate every time the water temperature drops below 60°C, therefore, whenever the cylinder loses heat and upon hot water draw off. For a 135 litre cylinder, a run time of about an hour (mathematically 47 minutes ignoring heat losses) would be sufficient for a 3 kW immersion heater. Of course the occupants may choose to ignore pasteurisation or, as was found in the EST heat pump trials, set the immersion temperature for 70°C to provide extra hot water as compensation for a small cylinder. Inadequate control of immersion heaters directly impacts on SPF_{H3} and SPF_{H4} efficiencies as well as on fuel bills.

Air source heat pumps and defrosting

MIS 3005 also requires that the designer include: “any energy required for defrost cycles”. No further details are given either within the MCS Installer Standards or in the Heat Pump Reference Materials*.* However, in presentation Spreadsheet 5[22] the

[21] Spreadsheet 5: http://www.screencast.com/t/s13i9Nrmp7 [accessed 23 September 2013]
[22] http://www.screencast.com/t/s13i9Nrmp7 [accessed 23 September 2013]

unsourced reference table used for sizing the heat pump and then assessing the SPF for annual energy use is recognised by the author[23] as taken from Daikin Altherma Technical Data (Daikin, EEDEN09-720 - 03/2009). Daikin present heat pump power output in two formats, "*Peak values*" and "*integrated values*", Figure 7-2, a difference that ranges from 11 to 14% between (-15) and 2°C ambient:

> "The integrated heating capacity and power input, is the average heating capacity and power input during 1 cycle (from end of defrost till end of the next defrost)."

Thus the selection is based on output including defrost and no additional calculations are required, although this is not made apparent in the webcast.

HEATING (Peak values)

Model	LWC	30		35		40		45		50	
	Tamb	HC	PI	HC	PI	HC	PI	HC	PI	HC	PI
006	-15	3.93	1.48	3.67	1.59	3.47	1.71	3.33	1.84	3.25	1.99
	-10	4.65	1.52	4.32	1.65	4.07	1.79	3.89	1.94	3.78	2.10
	-7	5.14	1.54	4.77	1.68	4.49	1.83	4.28	1.99	4.15	2.16
	-2	6.06	1.57	5.62	1.72	5.28	1.88	5.03	2.06	4.87	2.25
	2	6.89	1.57	6.38	1.74	6.00	1.91	5.72	2.11	5.53	2.31
	7	8.03	1.57	7.45	1.75	7.00	1.94	6.68	2.15	6.47	2.37

HEATING (integrated values*)

Model	LWC	30		35		40		45		50	
	Tamb	HC	PI	HC	PI	HC	PI	HC	PI	HC	PI
006	-15	3.50	1.40	3.27	1.51	3.09	1.62	2.97	1.75	2.89	1.89
	-10	4.14	1.45	3.85	1.56	3.62	1.70	3.46	1.84	3.36	2.00
	-7	4.52	1.45	4.20	1.58	3.95	1.72	3.77	1.87	3.65	2.03
	-2	5.27	1.46	4.89	1.60	4.59	1.75	4.38	1.92	4.24	2.10
	2	5.92	1.45	5.49	1.60	5.16	1.76	4.92	1.94	4.76	2.13
	7	8.03	1.57	7.45	1.75	7.00	1.94	6.68	2.15	6.47	2.37

* The integrated heating capacity and power input, is the average heating capacity and power input during 1 cycle. (from end of defrost till end of the next defrost).

Figure 7—2 Heat pump power sizing tables (Daikin Altherma, 2009)

To estimate heat pump output at design conditions and annual energy use, the designer must be able to interrogate tables such as Figure 7-2 or Figure 7-3, the same type of information but in graphical form. Where space heating and domestic hot water are at different temperatures, this will require two heat pump outputs to reflect the design outdoor temperature and the flow temperature concerned. For example, applying integrated values from Figure 7-2 to an air source unit in London, design outside temperature (-1.8)°C supplying underfloor heating at 35°C and domestic hot water at 50°C, the unit has outputs of 4.89 and 4.24 kW respectively. For the same conditions, the ground source unit in Figure 7-3, with a ground loop return temperature to the heat pump of 0°C, will provide 5.3 and 4.8 kW.

[23] This lucky observation results from having tested a Daikin Altherma air source heat pump for domestic hot water production (and familiarity with its documentation) at the Barratt Green House, BRE Innovation Park, UK, see Chapter 3.

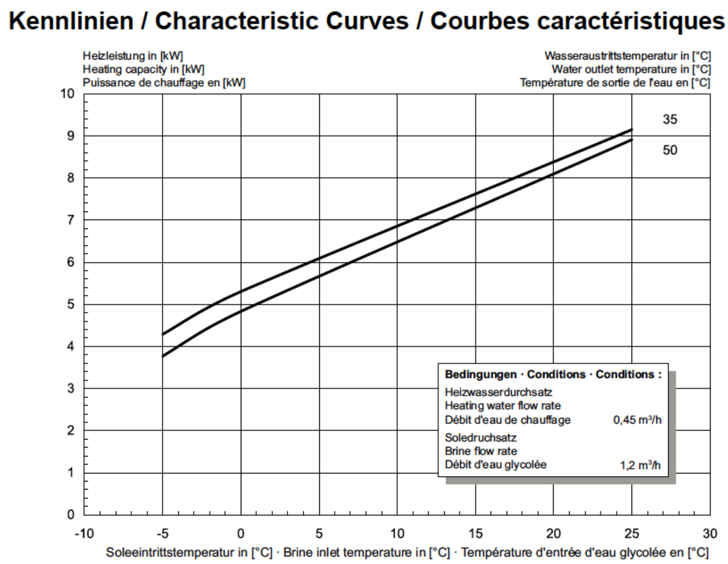

Figure 7—3 Heat pump power sizing tables (Dimplex GSHP SI range Ref: 452232.66.12)

Remarks on heat pump sizing and annual energy use

Even for those with an Plumbing NVQ Level 2 qualification, on a three or four day course, the learning outcomes are formidable since there is no design requirement in plumbing qualifications at Level 2 (Ofqual[a], 2012) The NVQ Level 3 plumbing qualification does have a specific unit for space heating which includes design:

> "Understand and apply domestic central heating system installation, commissioning, service and maintenance techniques", where the learning outcomes state: "Be able to apply design techniques for central heating systems," (Ofqual[b], 2012).

The competency level, however, will depend on how trainers interpret this particular learning outcome, their own level of knowledge and the methods used to assess its correct application. Personal experience of teaching and managing plumbing training indicates that the extent to which calculations are taught and, more critically, are understood, will range considerably from one training centre to another.

MCS have attempted to present all of the heat loss design and hot water criteria in a user-friendly manner, yet with a high technical content. There is no doubt that those who understand the processes outlined will succeed in sizing heat pumps far more appropriately than has been observed in the EST Trials. We note, with some consternation, that these learning outcomes require a comprehensive understanding of heat transfer modelling aligned to the interpretation of manufacturers' data; no little challenge for those with no formal training in building physics and heating calculations.

Sizing ground loop length

The MCS heat loss spreadsheets result in heat pump power (kW) and annual energy demand (kWh). The aim of ground loop sizing is to match heat extraction from the ground to heating demand at minimum ground loop circulator power. Chapter 6 of this thesis indicates a poor correlation between ground loop length and heat pump output. In response to these failings and to facilitate the ground loop design process, MCS provide reference materials[24] including, "Hydraulics design Pressure loss charts", "Hydraulics worksheets" and the "GSHP Hydraulics design guide: Procedure and charts for designing the hydraulics and associated pumping power of closed loop GSHP systems under MCS"[25].

The online training is aimed at design for horizontal ground loops, both straight and slinky, since both can be installed with standard construction machinery such as a JCB back-hoe. Boreholes are considered a specialist area since a borehole of, say 80 metres, will require a geological survey assessment for soil type and conductivity, detailed grouting procedures and a borehole contractor; such roles lie outside the context of the online training.

Heat transfer to the ground loop will depend on the ground mean temperature, the ground conductivity, the mean loop water temperature, the heat transfer characteristics of the pipe, the circulating fluid and the ground loop length. MIS 3005 Table 3 (p24), Figure 7-4, enables the designer to assess the required ground loop length through a series of 12 separate calculations.

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[24] http://www.microgenerationcertification.org/mcs-standards/reference-materials/heat-pump-reference-materials [accessed 23 September 2013]
[25] http://www.microgenerationcertification.org/images/GSHP_Hydraulics_Design_Guide_v1.0.pdf [accessed 23 September 2013]

Parameter	Value			Comments
Estimate of total heating energy consumption over a year for space heating and domestic hot water		kWh	[1]	(State calculation method)
HP heating capacity at 0°C ground return temperature and design emitter temperature, H		kW	[2]	
FLEQ run hours [1]/[2]		hrs	[3]	
Estimated average ground temperature		°C	[4]	
Estimated ground thermal conductivity		W/mK	[5]	
Maximum power to be extracted per unit length of borehole, horizontal or slinky ground heat exchanger (from the charts and look-up tables), g		W/m	[6]	
Assumed heat pump SPF (from heat emitter guide)			[7]	
Maximum power extracted from the ground (i.e. the heat pump evaporator capacity) G = [2]*1000*(1 – (1 /[7]))		W	[8]	
Length of ground heat exchanger calculated using the look-up tables L_b = [8]/[6]		m	[9]	(i.e. 2 no. 50m slinkies)
Borehole, horizontal loop or slinky spacing, d		m	[10]	
Total length of ground heat exchanger active elements, L_p = [9]*R_{pt}		m	[11]	(NB: does not include header pipes)
Total length of ground heat exchanger active elements installed in the ground, L_p'		m	[12]	(NB: state if proprietary software has been used to determine the design length)

Table 3 – Details of Ground Heat Exchanger design to be provided to the customer

Figure 7—4 MIS 3005, Table 3 - Details of ground heat exchanger design to be provided to the customer

Whilst ground loop sizing is logically based on balancing annual solar input to heat extraction, MCS state in MIS 3005 (2013 p20) that it should be based on keeping the ground loop return temperature above 0°C for 20 years and must, therefore, be subject to a limited maximum heat extraction rate from the surrounding soil. The calculation is based on the power, design step 1, and the energy demand, step 2, to provide annual "*full load equivalent hours"* (FLEQ), step 3, Equation 7-1:

$$\boldsymbol{FLEQ\ run\ hours = \frac{Total\ heating\ energy\ consumption}{Heat\ pump\ capacity} = \frac{kWh}{kW} = hours}$$

Equation 7-1

We should note that, in accordance with MCS training, the heat pump capacity is based on the space heating load (considered to far out-weigh that of DHW) and energy consumption is the combined space heating and DHW loads, minus any immersion backup.

The FLEQ equation, the basis of ground loop design, emphasises the importance of the designer's assessment of annual energy use (kWh), a requirement not specifically demanded of any other domestic central heating design whether gas, oil or solid fuel, since assessment of annual consumption has no impact on the actual efficiency of the system. In essence, it does not matter whether the designer gets annual consumption right or wrong when designing for other fuels - it does for heat pumps.

Mean ground temperature, step 4, is assumed to be the annual mean air temperature. Estimated ground conductivity, step 5, will require a soil analysis at the trench depth to assess soil type and moisture content. Soil conductivity is heavily influenced by water content and this will be dependent on time of year since the water table is dynamic in its level and flow rate, depending on weather conditions and ground slope. The decision on design conductivity (W/mK) is therefore a 'guesstimate', but at least one determined on the physics of heat transfer rather than some of the 'rules of thumb' noted in Chapter 6. The rate of heat transfer (W/m) is then read from charts based on FLEQ hours, type of loop, soil conductivity and the ground loop pipe diameter, step 6.

Next the designer must state: "Assumed heat pump SPF (from Heat Emitter Guide)", step 7. The assumed SPF impacts on the "heat pump evaporator capacity", G, step 8, "where H is the heat pump heating capacity determined in section 4.2.1c" [the space heating load].

$$G = H\left(1 - \frac{1}{SPF}\right)$$ Equation 7-2

We have seen that where separate temperatures are used for space heating and domestic hot water, the "Heat emitter guide" will provide two different SPFs, one high for low temperature emitters and one low for high temperature DHW. Since the DHW load is assumed to be small relative to that of space heating, it is the space heating "likely SPF" which is chosen.

Equation 7-2 calculates the heat extracted from the ground as an inverse function of SPF. This presents the designer with an output that provides a longer ground loop for a high SPF and a shorter loop for a low SPF, best explained with reference to the vapour compression cycle and the derivation of heat pump COP. An example is in order based on MIS 3005 Table 3 steps (1) - (12), shown as Figure 6-3:

For a 30,000 kWh annual energy consumption (1), 10 kW heating capacity at 0°C (2), the FLEQ running hours are 3,000 (3). At a ground temperature of 11°C (4), a ground conductivity of 1.5 W/mK (5), a horizontal loop will extract 10 W/m (6).

For a space heating flow temperature to radiators of 40°C, the Emitter guide provides a ground source "likely space heating SPF" of 4.1, therefore, G = 10 x 1000(1-1/4.1) = 7561 W. That is, 7561 W are extracted from the ground, combined with 2439 W from the compressor to achieve an output of (7561 + 2439) = 10 kW. The COP = 10/2.439 = 4.1, the "likely space heating SPF".

In comparison, for a space heating flow of 50°C, the Emitter guide provides an SPF of 3.4 and therefore G = 7059 W extracted from the ground. Since the heating requirement is 10,000 W, the compressor must now work harder to supply the additional 2941 W. The resulting COP = 10/2.941 = 3.4, the "likely space heating SPF".

The "*active*" ground loop length is, in effect, a source of energy and analogous to fuel for an oil or gas boiler. Heat pump output is the combination of this heat from the ground loop (Qin) and heat from the compressor (Win).

Since the flow and return temperatures provide the "likely space heating SPF" (MCS Heat Emitter Guide), the lower the SPF the more the heat pump relies on heat derived from the compressor. Thus, following MCS design guidance; a system with a low SPF will require a shorter ground loop than one with a high SPF.

The advantage of this sizing technique is that the ground loop is economically designed – it is as short as possible. The obvious drawback is that it reduces the amount of heat capable of being extracted from the ground should the building or heating system undergo subsequent retrofitting.

The resulting ground loop heat extraction (G), at an SPF of 4.1, in our example 7561 W (8), is divided by the heat transfer rate (W/m), assessed from "Ground loop sizing tables"[26] based on soil analysis and loop type, to provide the horizontal trench length.

At 10 W/m (6) the example requires 756 metres of trench for a horizontal straight loop (9). 756 metres is ¾ of a kilometre and therefore would generally be installed in parallel trenches where the minimum spacing is 0.75 m (10).

Where a slinky is specified the rate of extraction is higher per unit trench length and under the same FLEQ, soil and ground temperature conditions the extraction rate rises to 27 W/m length of trench. Since slinkies are loops, for a standard roll of 900 mm diameter, the effective length of pipe per metre of trench with no overlapping is approximately ($\pi d + 1$) and provides 3.8 metres per metre length of trench.

At 27 W/m, the trench length is 7561/27 = 280 metres (9). The shorter slinky trenches, extracting more heat per metre, will require a wider spacing of 3 metres (10).

The actual slinky pipe length required is calculated by multiplication of trench length by R_{pt}, where R_{pt} for slinkies ≥ 4. Thus the minimum slinky length is 280 x 4 = 1120 metres (11). Finally, MIS 3005, Table 3 requires the actual length of active ground loop installed (12).

Where ground conditions are found to not resemble the design assumptions, MIS 3005, p23 demands that:

> "should the geological situation on drilling or digging show substantial deviation from the conditions used in design or should drilling conditions become unstable or for some other reason the target depth or area not be achieved, the design of the ground heat exchanger shall be recalculated and the installation revised or adjusted if necessary."

The requirement is to assess site conditions during excavation and to be ready to re-design.

[26] http://www.microgenerationcertification.org/images/MIS_3005_Supplementary_Information_1_-_MCS_022_-_Ground_loop_sizing_tables_2011-09-02_v1.0.pdf [accessed 23 September 2013]

Remarks on sizing ground loop length

The analysis of the MIS 3005 method shows, yet again, that the designer must understand a range of technical issues that are outside the normal scope of those involved with gas or oil boiler installation. Apart from the assessment of annual energy demand for space heating and domestic hot water, required for FLEQ, the designer needs to make a series of assessments about ground conditions in order to provide the client with a tender document that includes design specification and price for the ground loop with the proviso of variation based on soil conditions encountered.

The sizing example applied to MIS 3005 Table 3, the 12 stage process, is based on relatively warm ground conditions and mid-table conductivities whilst the design heat loss of 10 kW and annual energy use of 30,000 kWh are not un-typical of existing poorly insulated housing and DHW load, results in either straight pipe of 756 m or slinky of 1120 m. For the same power (10 kW) and half the annual energy load (15,000 kWh) the lengths are 445 m or 552 m respectively. The EST Trial, as noted in Chapter 6, provides a limited selection of ground loop lengths but for four 11 kW and 12 kW heat pump units, where the slinky lengths range from 150 to 650 metres, some are almost certainly undersized and go some way to explain the relatively low seasonal performances encountered. That there could be significant undersizing is not surprising when comparing these slinky lengths with advice from 'Stafford Save Your Energy'[27], a project set up by Stafford Borough, UK with the support of various low energy advisory services and an energy supply company:

> "an average-sized house would need a loop of around 200 m, equivalent to a slinky in a trench 50-70 m long".

Ground loop diameter and pump specification

Heat transfer to the ground loop fluid is subject to the Reynold's number (Re), as is the flow resistance per unit length. For effective heat exchange the fluid velocity must be increased from laminar to turbulent flow to break up the stagnant "boundary layer" to improve heat transfer through the pipe wall into the circulating fluid. The Reynold's number is an assessment of flow regime where $Re = \rho Vd/\mu$, and where ρ is density, V velocity, d pipe diameter and μ viscosity. Both viscosity and density are dependent on the "*brine*" water/antifreeze mix. Turbulent flow results in a high Reynolds number, however, an increase in turbulence will also increase frictional resistance as the smooth flow over the boundary layer is disturbed to reveal the pipe wall roughness.

[27] http://www.staffordarea.saveyourenergy.org.uk/what/heating/gshp [accessed 23 September 2013]

The simplest expression of flow resistance is the Darcy equation where pressure drop in metres head (hf) = $fLV^2/2gd$, and where L is pipe length, V velocity, g gravitational acceleration and d pipe diameter. The function *f*, Darcy's friction factor, is found with a Moody diagram and is itself a function of the Reynold's number and the pipe wall roughness. MIS 3005 states that the Reynolds number in the ground loop "should be ≥ 2500 at all times" yet also requires that:

> "For all installations, the hydraulic layout of the ground loop system shall be such that the overall closed-loop ground collector system pumping power at the lowest operating temperature is less than 3% of the heat pump heating capacity" [previously 2.5% in MIS 3005 Issue 3.1a].

The "experienced worker", encountered at the short-course training session, is now exposed to logic of fluid mechanics and matching flow rate and pressure drop within a set of design criteria. The maximum ground loop circulator power for the 10 kW heat pump example, must be ≤ 300 Watts. The designer must be able to assess the friction losses through the entire ground loop, that is, the evaporator heat exchanger, the header pipes, fittings and valves and finally the horizontal or slinky "*active elements*" of the loop. MCS reference materials provide the "GSHP Hydraulics Design Guide v1.0", or to give it its full title, "Procedure and charts for designing the hydraulics and associated pumping power of closed loop GSHP systems under MCS" (GeoEnergy, 2012).

The power requirements for the ground loop circulator are dependent on the heat pump manufacturer's flow rate, the viscosity of the brine (itself dependent on the anti-freeze mix), the velocity and pipe diameter; that is, the total mass flow rate of brine through the index circuit - that circuit with the greatest pressure drop. For parallel loops, the index circuit is the length of any one loop, so coupling loops in parallel to the header manifold will reduce pump power demand. The elements of the three sections of a ground loop are shown in Figure 7-5 where "*header pipework*" will include fittings and valves.

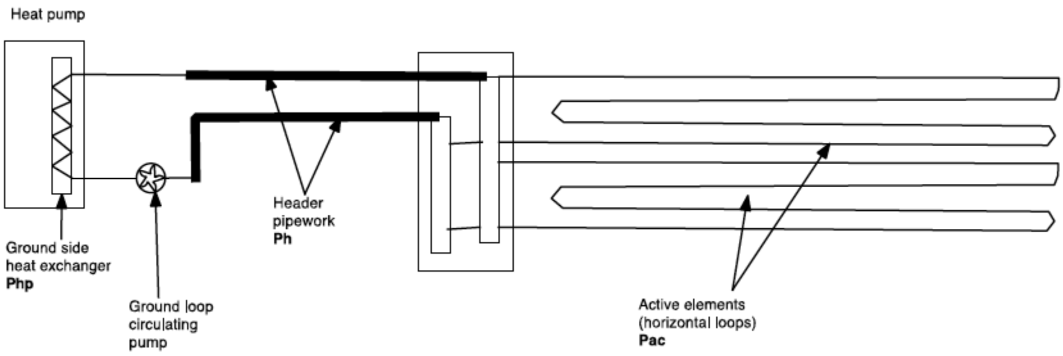


Figure 7—5 Basic hydraulic elements of closed loop GSHP (MCS, GeoEnergy, 2012 p5)

The MCS GeoEnergy guide provides a design methodology with pressure loss tables for both the active element and header pipes. The designer will need to interpret both the sizing tables (litres/second and kPa/metre of pipe) and manufacturer's data for heat exchange, pressure drop in the evaporator and size of manifolds. GeoEnergy provide the following total loop pressure drop equations based on Figure 7-5: The total permissible pressure drop (PPD) must be ≤ 3% of the heat pump output. The pressure drop, PD = 1.15 (Php + Ph + Pac) where 1.15 represents a 15% allowance for fittings and bends. Thus PD ≤ PPD.

The calculation procedure is provided in a series of algorithms, pages 10 to 12, based on normal or high efficiency pump selection, where a circulator pump has a normal efficiency of about 30% and a high efficiency pump of about 50%. Sizing is based on this initial choice and only on completion will the pump requirements become apparent and may then require a re-iteration of the process. Having understood this, the designer must select a suitable pump from a manufacturer where head and flow rate match the design criteria at less than 3%. Pump manufacturers generally specify their products in pressure units of metres head or kPa and flow rate in m^3/h or litres/s. All units must now be converted to a common base for pump selection calculations. The principles of the sizing algorithm are shown in Figure 7-6.

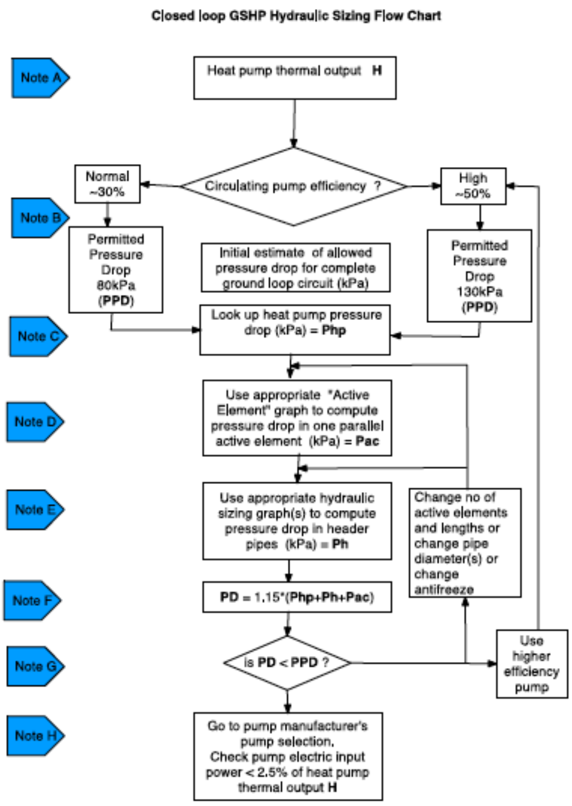


Figure 7—6 Partial closed loop GSHP Hydraulic Sizing Flow Chart (GeoEnergy, 2012 p10)

Pump sizing is based on pressure loss and flow rate. Within a pumped system of parallel pipework, typical of both UK two-pipe central heating design and ground loops, the maximum frictional losses, as previously stated, generally occur on the longest pipe run from the circulation pump, the index circuit. If the pump can overcome the pressure losses to the index circuit then it will be able to overcome all resistances in parallel circuits. The pump is then sized based on the index circuit pressure drop and maximum flow rate, that is, the total flow rate in the system. For a ground loop system, it is therefore only necessary to calculate the pressure drop in one parallel loop to add to that in the manifold and heat pump to estimate the total pressure loss in a system whilst supplying the total design flow rate.

To continue the use of our horizontal straight pipe ground loop example, the active element is 756 metres long and if designed as in Figure 7-5 has two parallel loops of 387 metres, requiring 8 trenches, each of which is 95 metres long. At 0.75 metres spacing the width is 5.25 metres providing an area of approximately 500 square metres.

Let us consider the Dimplex SI GSHP range[28], Figure 7-7. For a 10 kW output select the SI 11TE unit.

Device information for brine-to-water heat pumps for heating purposes SI 5TE - SI 11TE

1	Type and order code			SI 5TE		SI 7TE		SI 9TE		SI 11TE	
2	Design										
2.1	Degree of protection according to EN 60 529			IP 20		IP 20		IP 20		IP 20	
2.2	Installation Location			Indoors		Indoors		Indoors		Indoors	
3	Performance data										
3.1	Operating temperature limits:										
	Heating water flow		°C	Up to 58		Up to 58		Up to 58		Up to 58	
	Brine (heat source)		°C	-5 to +25		-5 to +25		-5 to +25		-5 to +25	
	Antifreeze			Mono-ethylene glycol		Mono-ethylene glycol		Mono-ethylene glycol		Mono-ethylene glycol	
	Minimum brine concentration (-13 °C freezing temperature)			25%		25%		25%		25%	
3.2	Temperature spread of heating water (flow/return flow)	at B0 / W35	K	10.1	5.0	9.9	5.0	10.5	5.0	10.1	5.0
3.3	Heat output / COP	at B-5 / W55 [1]	kW / ---	3.8 / 2.0		5.6 / 2.2		7.7 / 2.3		9.4 / 2.4	
		at B0 / W45 [1]	kW / ---		5.0 / 2.9		6.6 / 3.0		8.7 / 3.2		11.2 / 3.2
		at B0 / W50 [1]	kW / ---	4.8 / 2.8		6.7 / 2.9		9.0 / 3.1		11.3 / 3.0	
		at B0 / W35 [1]	kW / ---	5.3 / 4.3	5.2 / 4.1	6.9 / 4.3	6.8 / 4.1	9.2 / 4.4	9.0 / 4.2	11.8 / 4.4	11.7 / 4.2
3.4	Sound power level		dB(A)	54		55		56		56	
3.5	Heating water flow with an internal pressure differential of		m³/h / Pa	0.45 / 1900	0.9 / 7400	0.6 / 3300	1.2 / 13000	0.75 / 2300	1.6 / 10300	1.0 / 4100	2.0 / 16100
3.6	Brine throughput with an internal pressure differential (heat source) of		m³/h / Pa	1.2 / 16000	1.2 / 16000	1.7 / 29500	1.6 / 26500	2.3 / 25000	2.2 / 23000	3.0 / 24000	2.7 / 20000
3.7	Refrigerant; total filling weight		type / kg	R407C / 1.2		R407C / 1.1		R407C / 1.6		R407C / 1.7	

Figure 7—7 Dimplex SI GSHP, p12

Note that the brine temperature range is (-5) to 25°C, the antifreeze specified as mono-ethylene glycol and that the manufacturer suggests an anti-freeze protection of (-13)°C for a 25% anti-freeze mix. At 0°C/35°C (EN 14511) temperatures and a 5K space heating flow and return

[28] http://www.dimplex.de/downloads/uploads/si5-21te_fd8611_gb-2007-05-15.pdf [accessed 23 September 2013]

temperature difference, the maximum heat output is 11.7 kW with a minimum brine mass flow rate 2.7 m^3/h and internal pressure drop of 20,000 Pa.

Convert all units:

Brine flow rate: 2.7 m^3/h = 2.7 x 1000 (l/m^3)/3600 (s/h) = 0.75 litres/second

Brine throughput with an internal pressure difference: 20,000 Pa = 20,000/1000 (Pa/kPa) = 20 kPa

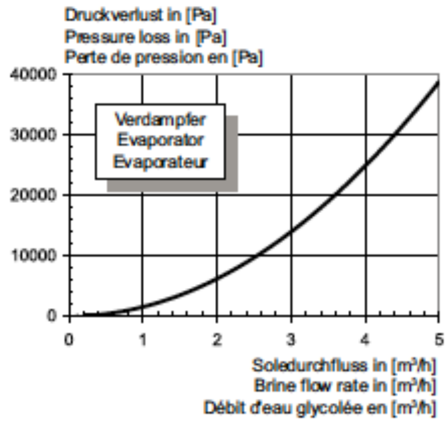


Figure 7—8 Dimplex SI 11TE Evaporator pressure drop graph, p2.4

Note that whilst Figure 7-7 shows the "*internal pressure difference*" to be 20,000 Pa for a 2.7 m^3/h flow rate, Figure 7-8 shows the evaporator pressure drop to be closer to 11,000 Pa. We can only err on the high side, assuming that the "*internal pressure difference*" includes brine circuit manifold connections to the active elements, Figure 7-9, and hence the correct value for *Php.*

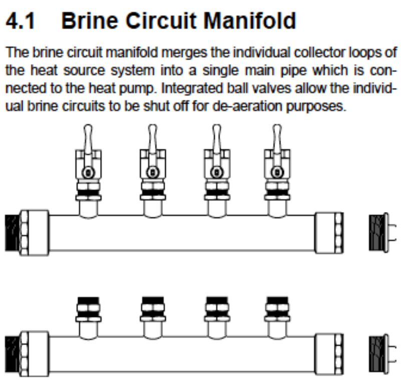


Figure 7—9 Dimplex SI 11TE Manifolds, p4

Next select the appropriate "active element pressure drop chart" (GeoEnergy, 2012 pp22-29) to calculate pressure loss per metre run based on the manufacturer's "minimum brine concentration (-13°C freezing temperature)" mono-ethylene glycol [ethylene glycol] and two choices of polyethylene pipe, SDR 11 and SDR 17.

SDR is the 'standard dimensional ratio' of the pipe outside diameter to wall thickness. SRD 17, in comparison to SDR 11, has a thinner wall and therefore a slightly bigger bore; it holds more water per nominal size and, for the same flow rate, has less resistance per metre run. Note that the ground heat extraction tables, previously used to calculate the loop length, are based on 25 mm diameter SDR 11 pipe. Since all larger diameter pipes have a greater surface area, we will assume at least the same heat transfer properties. Select SDR 11, Ethylene glycol, Figure 7-10 (GeoEnergy, 2012 p31)

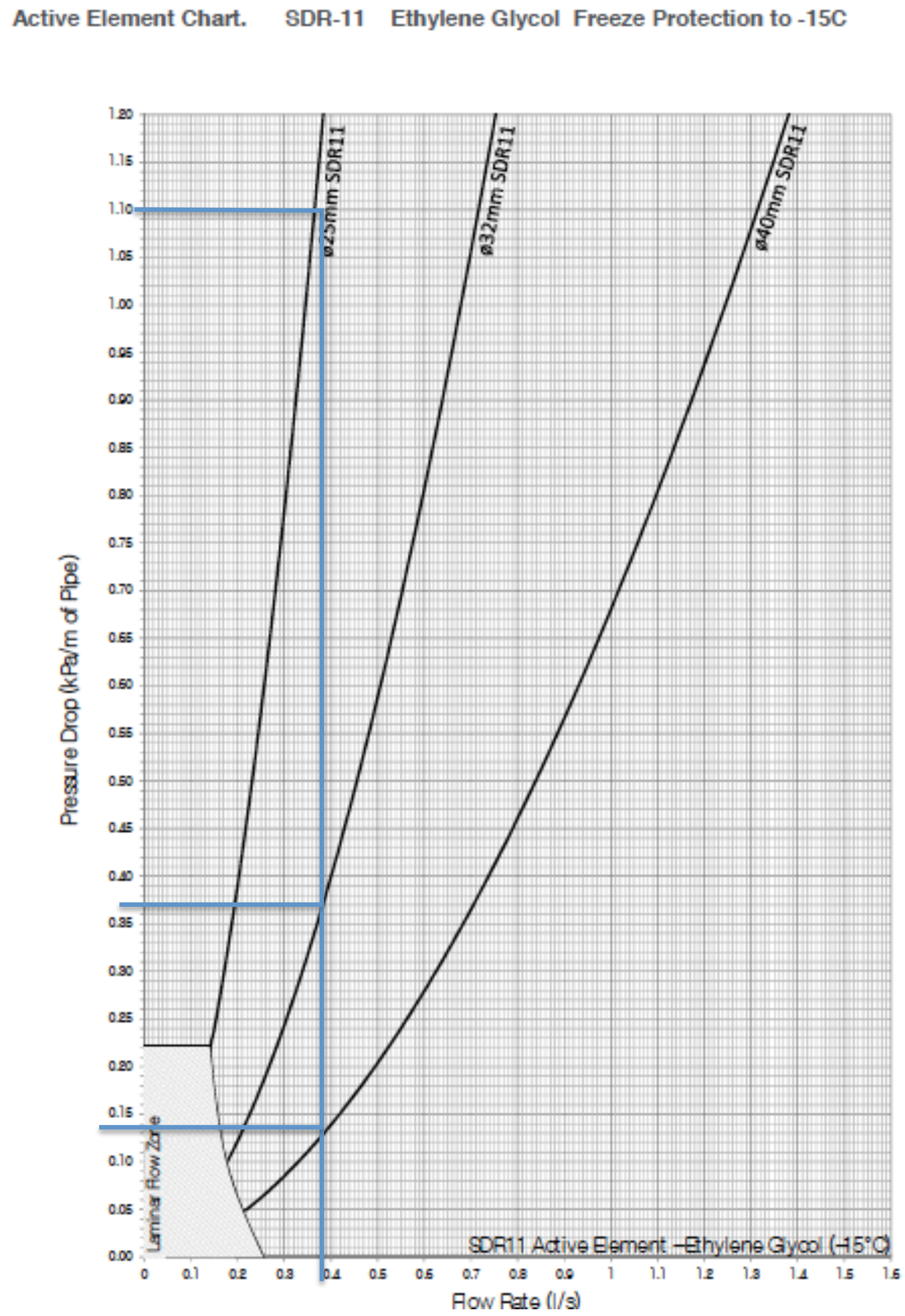


Figure 7—10 Active element pipe sizing graph showing sizing example at 0.375 litres/s (GeoEnergy, 2012 p31)

For a single loop at a flow rate of 0.75 litres/s the pressure drop is approximately 0.41 kPa/metre for 40 mm diameter SDR 11 polypropylene pipe; Pac = 0.41 kPa/m x 756 m = 310 kPa.

For a two loop system, the flow rate per loop is 0.375 litres/s and 40mm diameter, the pressure drop reduces to 0.125 kPa/m, or 0.125 kPa/m x 378 m = 47 kPa.

Reducing the pipe size to 32 mm increases the pressure drop to 0.36 kPa/metre. Hence Pac = 0.36 kPa/m x 378 m = 136 kPa. For two loops, the pressure drop across the heat pump (20 kPa) and active element (136 kPa) is 156 kPa and will require a re-design.

The options are to change pipe, change antifreeze mix or to increase the number of parallel loops. The process is iterative; it will depend on land availability and must, at some point, consider the additional labour, materials and machine hire costs associated with increasing the number of loops and thus the number of trenches. Such an iteration is shown in Table 7-1.

No of loops	Trench length	Pipe length	Flow rate	Pipe dia	dP kPa/m	dP Total	Comments
1	756	756	0.75	40	0.41	310	dP too high
2	378	378	0.375	40	0.125	47	
2	378	378	0.375	32	0.36	136	dP too high
3	252	252	0.25	40	0.065	16	
3	252	252	0.25	32	0.18	45	
3	252	252	0.25	25	0.59	149	dP too high
4	189	189	0.1875	25	0.36	68	Laminar flow

Table 7—1 Ground loop design options for pump selection

Selecting 3 loops at 40 mm diameter provides a flow rate of 0.25 l/s and a pressure drop of 0.065 kPa/m. The selection runs across the laminar flow zone and to ensure that turbulent flow is being achieved it is worth checking the Reynolds number.

Reynold’s number checking: Re = $\rho Vd/\mu$

For 40 mm pipe, the inside diameter can be found from SDR = outside diameter/wall width.

Inside diameter = 0.0364 m (36.4 mm)

The continuity equation ($m^3/s = m^2 \times m/s$) provides velocity:

$m^3/s = 2.7(m^3/h)/3600(s/h) = 0.00075\ m^3/s$

Area = $\pi d^2/4 = \pi \times 0.0364^2/4 = 0.00033124\ m^2$.

V = 0.00075/0.00033124 = 2.264 m/s

Density = 1051 kg/m3 and viscosity = 0.00341 Pa.s (GeoEnergy, 2012)

Re = (1051 x 2.264 x 0.0364)/0.00341

Re = 25400 > 2500

Clearly the laminar zone is only within the bottom-left grey region of the graph.

The final step is to consider the header pipework. GeoEnergy provide only two header pipe charts for SDR 11 and SDR 17, both ethylene glycol for (-10)°C. The length of the header pipework is entirely dependent on the distance between the heat pump and the manifolds. If we assume a combined 6 metres flow and return, then the pressure drop at 0.75 litres/s and 40 mm diameter is 0.41 kPa/m x 6m = 2.46 kPa.

Pump selection based on 40 mm pipework for the heat pump, two active elements and headers:

PD = 1.15(Php + Pac + Ph)

PD = 1.15(20 + 47 + 3)

PD = 80.5 kPa

Pump selection = 0.75 litres/second (2.7 m^3/h) at 81 kPa.

Alternatively, change the active elements to three loops at 32 mm for a saving in pipe cost although an increase in labour and machine hire:

PD = 1.15(20 + 45 + 3)

PD = 78 kPa

Find a pump

Select the Grundfos Magna 50-100F circulating pump[29], low energy consumption energy class "A". For a two active element installation at 40mm, the system design is for 0.75 l/s at 81 kPa. Flow rate setting 'flow control valves' should be installed on all active element loops and on the header pipes for commissioning when the pump speed is set to provide the design flow rate, Figure 7-11.

Figure 7—11 Flow setting valve

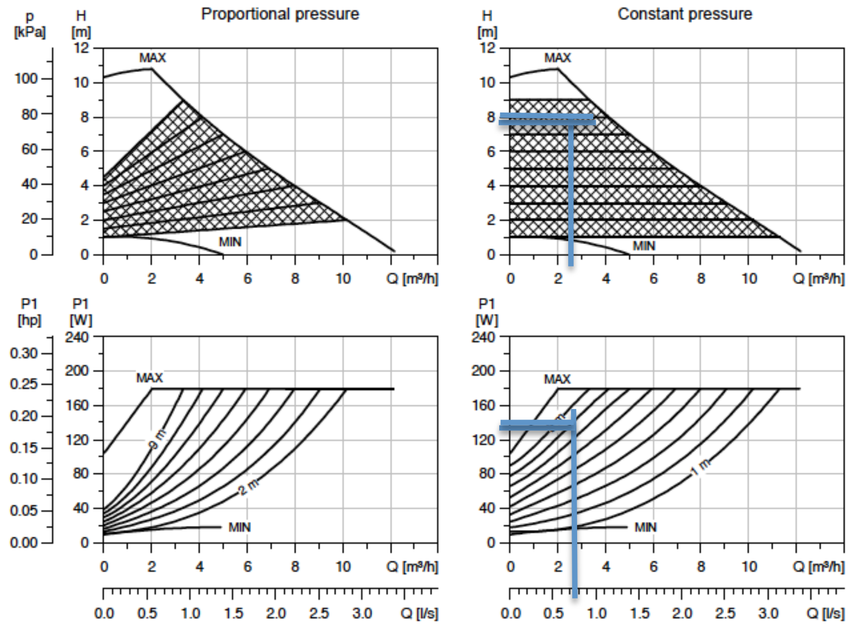
The pump is set to constant pressure, Speed 8 or 9, Figure 7-12. The power required is approximately 140 Watts or approximately half the design maximum (1.4% < 3%). Since the

[29] http://net.grundfos.com/doc/webnet/magna/downloads/Magna_databooklet.pdf [accessed 23 September 2013]

pressure drop is slightly over 80 kPA the speed setting should be 9, however, the fittings allowance of 15% may be excessive and speed 8 sufficient.

Alternatively, for three 32 mm loops and pump settings of 0.75 l/s and 78 kPa, Speed 8, the pump requires slightly less than 140 Watts.

MAGNA 50-100 F



Electrical data

U_n [V]		P_1 [W]	$I_{1/1}$ [A]
1 x 230-240 V	Min.	10	0.1
	Max.	180	1.26

Figure 7—12 Grundfos Magna 50-100F pump: flow rate, head and power

Both 32 and 40 mm active element options fall within the 3% rule; the final decision is almost certainly based on prime cost for the contractor. It should be noted that once the pressure drop rises to over 100 kPa, it is difficult to find an appropriate pump, as can be seen from the Magna 32-120F, where the power demand is greater than 300 Watts, Figure 7-13.

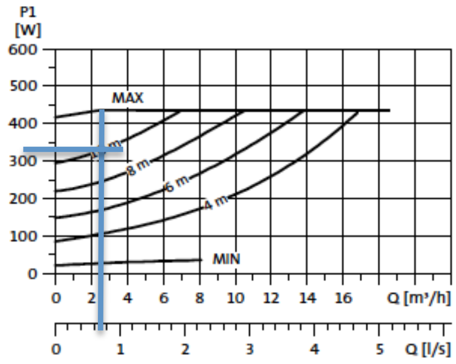


Figure 7—13 Grundfos Magna 32-120F pump

The design is complete; it now requires the formal presentation of the results in the "MCS GSHP Hydraulics Worksheet"[30], Table 7-2. The designer must provide evidence of having carried out the procedure.

However, it may be necessary to recalculate the entire process:

> "should the geological situation on drilling or digging show substantial deviation from the conditions used in design or should drilling conditions become unstable or for some other reason the target depth or area not be achieved, the design of the ground heat exchanger shall be recalculated and the installation revised or adjusted if necessary", (MIS 3005 clause 4.2.16, p16).

[30] http://www.microgenerationcertification.org/mcs-standards/reference-materials/heat-pump-reference-materials [accessed 23 September 2013]

Project Name	Gleeson				
Antifreeze type	Ethylene glycol		Freeze protection	(-15)	(°C)
Heat pump information					
Make/Model	Dimplex SI 11TE		Ground flow rate	0.8	(l/s)
Thermal output at B0/W35	11.7	**H** (Watts)	Pressure Drop **Php**	20	(kPa)
Active Element Information					
Pipe OD	40	(mm)	No.of active elements 2		
			Flow rate per element	0.4	(l/s)
Pipe Type	SDR 11		Pressure Drop **Pac**	47	(kPa)
Header Information					
Pipe OD	40	(mm)	No. of pairs headers 1		
			Header flow rate per pair	0.8	(l/s)
Pipe Type	SDR 11		Pressure Drop **Ph**	3	(kPa)
			Sub total **Php+Pac+Ph**	70	(kPa)
			x 1.15 for fittings etc	81	**TPD**
Pump Selection					
Make / Model	Grundfos Magna 50-100F		Speed setting	8	
Electrical input at duty point of **F** (l/s) and **TPD** (kPa)	140	Watts	=	1.4	% of **H**

Table 7—2 MCS GSHP Hydraulics Worksheet. Example for two active elements at 40mm diameter

Ground loop layout

The charts provide the length for each parallel ground loop, the designer will now have to design a layout. The permutations for layout will, in practice, be dependent on the area and shape of land available and the type and cost of trenching, its width and depth. Pipes may be buried using a variety of machinery from the industry-standard back-hoe, typically a JCB, or specialist plant such as a chain trencher which results in minimal ground disturbance and back-filling, Figure 7-14.

Figure 7—14 Chain trencher. Image from Ice Energy[31]

The loop may be continuous, with, as we have seen, the proviso that the frictional resistance of a single loop may be too high, or a set of parallel loops installed in a 'reverse return' design layout, where each loop is equidistant from the flow and return header pipes and therefore each loop has the same pressure drop as the index circuit; the system is theoretically self-balancing, Figure 7-15.

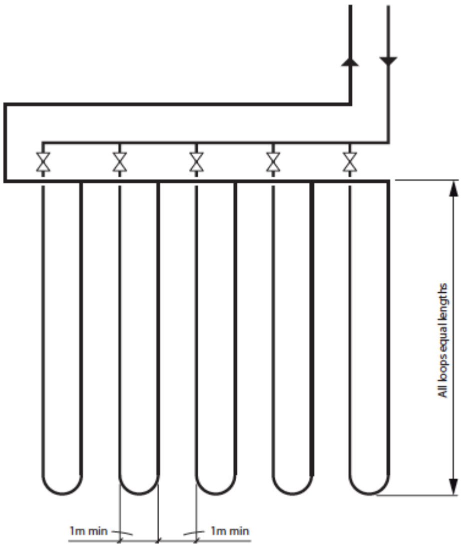


Figure 7—15 Parallel loops designed for reverse return. Image from NIBE[32]

[31] http://www.iceenergy.co.uk/Renewable-Energy-for-Homeowners/Renewable-Energy-Case-Studies/Macclesfield-Ground-Source-Heat-Pump-Solar-PV-Inst [accessed 23 September 2013]
[32] http://www.nibe.co.uk/Documents/nibe_co_uk/TIF_UK_inst-ground-collector.pdf [accessed 23 September 2013]

The design of the loops is itself an iterative process. Applying the process to Figure 7-5, where each loop has three turns and thus four trenches, the design iterations will be similar to Table 7-3.

Total length required	No of loops	Pipe diameter	Length of loop	No of trenches per loop	Total trenches	Spacing	Total width	Length of trenches
756	2	40	358	4	8	0.75	6	94.5
756	3	32	252	4	12	0.75	9	63

Table 7—3 Two options for ground loop layout

The sizing exercises provide some comparison with the EST trial data for ground loops. The Chapter 6 graphical analysis of horizontal ground loop length and heat pump output is reproduced in Figure 7-16 with the addition of the current example (756 m). We may infer from the linearity that there is at least consistency in the EST trial straight pipe ground loops. A very approximate rule of thumb arises for straight loops, based on Figure 7-16:

Pipe length = 70 m/kW heat pump output at 0°C/35°C.

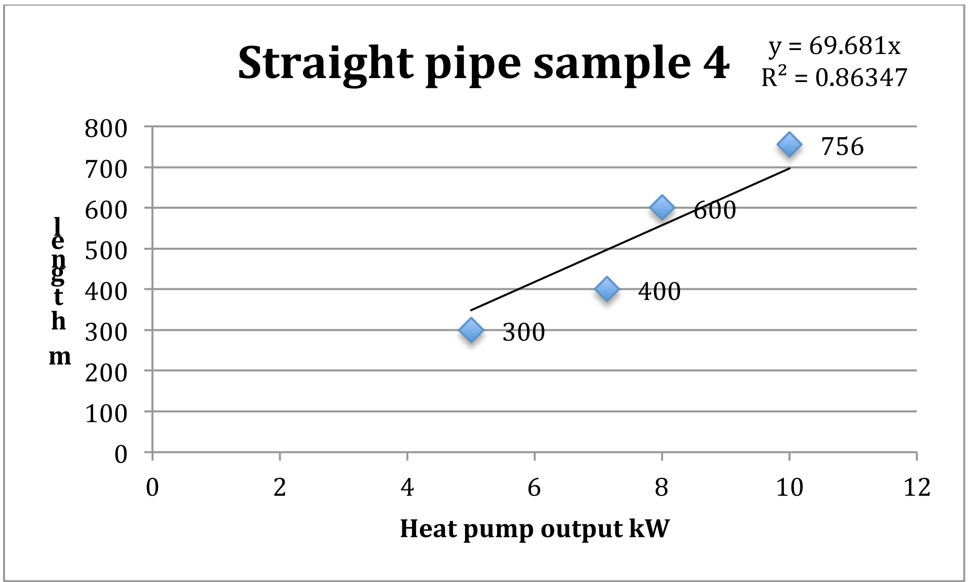


Figure 7—16 Tentative correlation between straight loop length and heat pump power

Remarks on ground loop and pump specification

The MCS webcasts do not address the design procedure for the ground loop beyond the calculation of minimum length. For whatever reason, perhaps one of time since the webcast series are based on the original DECC one day training programme, the final calculations for ground loop diameter and pump selection are covered only in MCS literature. To understand the ground loop design process and to tie it into circulating pump selection is highly technical, quite possibly an exercise for Engineering Council classified 'engineering technicians' (Eng Tech)[33]. The designer needs to be able to interrogate heat pump manufacturers' literature, convert all units to a common base, apply iterative sequences for pipe diameter selection and then understand enough about pump graphs and the relation between flow rate, pressure drop and power, to make an appropriate selection. Finally, they must present the evidence in the form of the Hydraulics worksheet. However, the designer must still fit the ground loops into the area of land available, requiring another iterative process which will depend on trench spacing, volume of trench spoil, availability of machinery and labour. These calculations are best suited to a spreadsheet format, again an issue of numeracy and ICT functionality.

Having analysed the requirements for calculating the length, the diameter and the ground pump selection, it is apparent that this particular exercise would be difficult to cover in a short course aimed at a cohort with no formal design education. The complexity of ground loop sizing, along with the cost in design time, machine hire, labour and materials is sufficient evidence to support air source heat pumps as the preferred option; an assumption reinforced by MCS registered installation statistics[34], where of the 23,053 heat pumps registered under the scheme (July 2009 - July 2013), only 24% are ground source.

Buffer vessels, tariffs and controls

MCS provides no guidance on the design of the heating system other than radiator sizing for lower temperatures and, having established the heat loss coefficient, the suitability of underfloor heating with different surface finishes. The daily volume of stored domestic hot water is part of the annual energy calculations and cylinder sizing support literature. With regard to cylinder primary coil design, MIS 3005 clause 4.2.4 states:

> "Domestic hot water heat exchangers for heat pump systems tend to require a much greater heat exchanger performance as compared to traditional combustion-based heat sources (i.e. boilers). For coil-type heat exchangers, this usually requires a significantly greater heat exchanger area," p17.

The hydraulics of the system, including the merits of buffer vessels, is not discussed.

[33] Engineering Council Eng Tech description: http://www.engc.org.uk/engtech.aspx [accessed 23 September 2013]
[34] http://www.microgenerationcertification.org/about-us/statistics [accessed 23 September 2013]

Since the scheme is designed for those in the domestic central heating industry, it must be assumed that they are expected to know how to pipe size for radiators and a cylinder, select a central heating pump and follow a manufacturer's installation instructions. Since many heat pumps are supplied with a heating circulating pump, and radiators are generally specified as the chosen emitter, under these circumstances, the actual installation is identical to any UK central heating system apart from the buffer vessel option and the influence of lower temperatures on radiator sizing. The increased mass flow rate $\left(\dot{m} = \frac{\dot{Q}}{\Delta T}\right)$ due to the smaller temperature difference across the radiator, generally 5K, is likely to join the 'fudge factors' to be overcome by sufficient pump head to increase the flow rate, albeit, at the expense of noise. Finally, no guidance is given as to the most appropriate combination of electricity tariff and control protocol including the fact that it is apparent that consumers will struggle with the majority of control settings.

EUCERT Heat Pump

From as early as 2002, the European Heat Pump News[35], mouthpiece of the European Heat Pump Association (EHPA), reported on the need to establish a comprehensive training programme:

> "The working programme of the group will include the collection and exchange of relevant information about national training schedules, regarding contents, organisational matters, time schedules, financial matters, etc. We plan to publish the information on the EHPA website. We are also developing a common quality standard, and discussions will soon be held about defining the contents of a standard training course." (EHPN, 2002)

The results are the EHPA training and certification program for heat pump installers, the EUCERT HP, described as an: "**Education and certification** programme for heat pump installers"[36]. Note the emphasis on 'education' rather than 'training' (bold in the original). The Education and Training committee have produced a programme for installers and are currently developing another programme aimed at ground loop design and installation through Geotrainet[37].

[35] http://www.pac.ch/dateien/EHPN_Issue_4=3.pdf [accessed 23 September 2013]
[36] http://www.qualicert-project.eu/fileadmin/Qualicert_Docs/Events/EUSEW11_EBC/20110412_EHPA_EUCERT.pdf [accessed 23 September 2013]
[37] http://www.geotrainet.eu/moodle/ [accessed 23 September 2013]

The EUCERT is based on a common "curriculum" for all training centres with a prescriptive workshop specification, specified as non-manufacturer specific. The programme is designed for 36 hours of delivery including 6 hours of practical training and final examination. The course content is both broadly educational and technology specific.

The EUCERT "Heat pump installer manual" (ConstructionSkills, UK) sets high educational expectations for its potential students. Each section of the manual is provided with self-assessment questions that clearly expect the student to have understood and be able to apply the information in the preceding section. For example, investment appraisal covers the topic from simple payback through to net present value and whole life costing. The self-assessment expects the student to provide an investment comparison based on different discount rates (p33). Within the technical sections, there is the same high expectation, evidenced through the discussion of cumulative run hours and the task of plotting hours against temperature, temperature against power, hours against power and finally "*Putting it all together*" (p196). One would presume that a person with no formal training would falter at such tasks and the EUCERT programme raises uncertainties regarding "the experienced worker route" and the technical and "functional skill" abilities of those with NVQ Level 2. The UK, which whilst a member of the EHPA Education and Training Committee and signed up to EUCERT through BSRIA, remains outside of the EUCERT programme, offering only MCS accreditation[38].

EUCERT appears to be influenced by the Continental model of VET, that is, the combination of what has been described as "occupational capacity" and "independent agency" with "work capacities" and "task-based skills" (Brockman, 2009). Occupational capacity may be described as the ability to see oneself, not through a skills-only definition (the ability to install heat pumps), but as part of a broader profession whose aims are to provide thermal comfort in an environmentally sustainable fashion, working alongside others as part of a built environment profession, with personal, technical and civic goals.

Occupational capacity is evident through the introductory chapters that focus on marketing and sales, costs and investment appraisal, environmental relevance of heat pumps, energy efficient buildings, thermal comfort and other renewable technologies. Thus the heat pump is placed in its social and environmental context before the student is treated to the technical intricacies of operation and design.

[38] http://www.ehpa.org/european-certified-hp-installer/participating-countries/ [accessed 23 September 2013]

The EUCERT fits that category of VET with a high general educational input and that distinguishes between *Fähigkeiten*, "the way one conducts oneself in the context of a range of activities" and *Fertigkeiten*, "the way in which one acts on the materials with which one is working". The combination of both abilities leads to a *berufliche Fähigkeit*, described by Brockman (Brockman et al, 2011) as an occupational capacity that integrates all knowledge, practical wisdom and understanding to practice an occupation.

Similar complex assignments of occupational capacity occur in France where *savoir* (knowledge), *savoir faire* (know-how) and *savoir être* (attitude) blend to provide a definition of competence far removed from the simple *can do* model exemplified by UK VET and evidenced by the short course training for MCS accreditation. Implicit, and at times explicit, in the MCS webcasts is the expectation that *this is complicated*, *you may not have done this kind of thing before*, and the warning in the webcasts of "*garbage in - garbage out".* MCS resources are designed for rote learning rather than critical thinking, a step-by-step guide to heat pump design; *do it this way and it will work.* Such different approaches to VET, the Anglo-Saxon "task based" versus the Continental "occupational capacity" resonate at European Union level with the current debates over the European Qualifications Framework[39] and agreement over knowledge, skills and the definition of competence, whether narrow or broad-based.
The ultimate test of success for UK MCS installer standards will be the efficiency of registered heat pump installations. Some measure of this success should be available through the UK Renewable Heat Premium Payment scheme (RHPP) where all installers are MCS registered and, as such, have provided evidence of, *inter alia*, suitable training. The RHPP website comments:

> "This scheme will also allow us to learn more about what people think of these technologies and *how they perform in a variety of conditions* [author's italics]." [40]

RHPP heat pumps have been monitored for SPF_{H2} results. At the time of writing (February 2013), whilst provisional results are in, they are being treated as confidential.

Summary

The aim of this chapter is to analyse the MCS guidance and online training for heat pump installers with reference to the EST trials and their short-fall in design practice. The MCS have responded with a comprehensive suite of documents, spreadsheets, online software and webcasts to cover heat loss calculations and annual energy consumption, heat pump and emitter selection. Providing that the process is understood, this part of the overall programme provides a sound basis for those designing and installing air source heat pumps. Whilst ground

[39] EQF: http://ec.europa.eu/eqf/home_en.htm [accessed 23 September 2013]
[40] https://www.gov.uk/renewable-heat-premium-payment-scheme [accessed 23 September 2013]

loop length is the subject of a webcast, pipe sizing and pump selection, vital to effective operation, is left to the designer to understand from the GeoEnergy document and hydraulic tables. For a qualified engineer, this is clearly achievable and reflects well on the quality of MCS documentation.

For the unqualified "experienced worker" and those with NVQ Level 2 or equivalent qualifications, the analysis proves the need for a well-developed general educational level in terms of "functional skills" alongside a comprehensive understanding of design criteria, nomenclature and engineering SI units. The MCS programme is still developing and, as was the case with UK Gas Safety training, will in the future demand more rigorous evidence of competence (Jones, 2013) through the assignment for each installation of various new roles such as a "nominated technical person" and "designer". From the above analysis of design knowledge and ability, this is likely to see the "nominated technical person" assigned 'duty holder' status for the whole installation. The resulting performance of the current Renewable Heat Premium Payment scheme and the proposed Renewable Heat Incentive[41] will, no doubt, drive the industry in this direction.

For fundamental reasons, the performance of heat pumps is much more sensitive to design and installation than the technologies that they replace, and, because they are a new technology for most installers, the failures associated with poor understanding of the engineering criteria impact directly on their seasonal performance, on the consumer and the general perception of the technology. In response, the heat pump industry has set standards that none of its equivalent competitors have to meet. Every installation has to be registered, supplied with a series of calculations that predict the annual running cost, the technical specification of the heat pump and emitters plus the design worksheets for ground source units. For renewable heat payments, under the RHI, each installation will also be monitored and its real efficiency available to the householder. None of these requirements apply to gas boilers.

Gas boilers, in contrast, are a mature market where manufacturers supply units that need little other than connection to the flue and pipework:

> "wall hung boilers dominate the UK market and within that sector combi boilers account for about 74% of the market. In Germany and France however, heat pumps have a much higher share of the boiler market" (Eljidi, 2011).

[41] http://www.rhincentive.co.uk/ [accessed 23 September 2013]

Gas boiler automatic modulation to control heating output, built-in controls and diagnostic software are commonplace and the boiler sufficiently robust to operate between 82 and 90% efficiency under virtually all circumstances (Micro CHP Accelerator, 2007). Heat pump manufacturers must endeavour to apply such automatic functions to all heat pumps. Manufacturers need to develop self-adaptive controls for fine-tuning weather compensation.

The research has shown that complex mechanical systems operating at small temperature gradients require a level of technical design ability that is more closely associated with formal professional engineering qualifications rather than the more installation-oriented programmes associated with UK vocational training at NVQ Levels 2 and 3. The imposition of such a demand on the UK domestic heating industry would require the development of a deeper educational input into mainstream VET along with control mechanisms such as individual, as opposed to company, registration of designers and installers, similar to that for gas installers. VET providers, both private and public, also need to be rigorously assessed in terms of staff technical ability and training provision.

Given the UK dominance of the combi boiler with its automated modulating fuel rate setting, the gas safety requirements for most central heating installations amount to soundness testing of the gas pipes, checking the position of a balance flue outlet and ensuring ventilation provision (Gas safety regulations, 1998). There is as yet no contractual requirement for a comprehensive set of design calculations and running costs. There is no registration for every installation beyond Building Regulation compliance, which demands that the installer be a member of a competent persons scheme[42]. With regard to technical development, regulation and ease of installation, there is no "level playing field" between gas boiler and electric heat pump. Heat pump manufacturers should recognise this, pursue rapid product research and development and offer installers design services. Improved knowledge from MCS training, supported by a technically competent supply chain will result in improved installation quality.

[42] http://www.competentperson.co.uk/ [accessed 23 September 2013]

Chapter 8 Discussion and conclusions

How well do heat pumps work in practice? To what extent are heat pump field trial results being communicated in a consistent fashion and what lessons can be learned from heat pump field trials?

Introduction

This research has grown out of an interest in heat pump performance based on an initial investigation into the operation of a specific heat pump installation. In 2009 I was invited, as a chartered building services engineer and potential PhD research student, to join a University College London investigation into the operation of an air source heat pump at the Barratt Green House in the Building Research Establishment's Innovation Park, UK, with a particular focus on domestic hot water production. This pilot project provided excellent efficiencies and invaluable experience of the 'trials and tribulations' of field research and the importance of understanding both the operation of the technology and the monitoring system. The results of the UK Energy Saving Trust's heat pump field trials were released in 2010 to chorus of misunderstandings regarding the basic functions of heat pumps and their applicability to residential heating. There appeared to be a mismatch between what had been witnessed at the Barratt Green House, in continental European field trials, and the UK trial results.

In apparent contradiction to the EST trials, both UK Government policy and EU Directives directly promote heat pumps as integral to meeting low carbon targets. At the European Union level, the Renewable Energy Sources Directive (EU, 2009) specifically identifies heat pumps in its Annex VII. Within the UK, both DECC and the Committee on Climate Change (CCC, 2010) foresee a key role for heat pumps in enabling the UK to fulfil its climate and energy goals. Among its many references to heat pumps the CCC report states:

> "Buildings: Direct emissions from heat in buildings are reduced significantly by 2030, as a result of major improvements in energy efficiency and roll-out of low-carbon heat, especially heat pumps," p29.

The UK Government's more recent document, "The Future of Heating: A strategic framework for low carbon heat in the UK" (DECC 2012), similarly enthuses over heat pumps:

> "As the electricity system decarbonises, technologies such as heat pumps and even electric resistive heating in buildings will be an increasingly effective way to decarbonise heat supply," p18.

> "Many new homes are now fitted with a heat pump, *able to operate three to four times more efficiently than a gas boiler*, and businesses are increasingly using heat pumps as a convenient way to both heat and cool their buildings," p39 (author's itialics).

> "[H]eating technologies that use low carbon electricity hold particular promise, especially as electricity is universally available and technologies here are relatively established. In addition, the high efficiencies of heat pumps, combined with improved building and storage technologies, could counteract the relatively high costs of electricity, making electrical heating an affordable option, particularly if the manufacturing and installation costs of heat pumps come down as volumes increase," p44.

> "In suburban and rural areas, in particular, low carbon heating technologies at the level of individual buildings will be necessary. Here, heat pumps are expected to provide substantial quantities of heat where heat networks … are not technically or economically viable," p56.

> "The Government's vision is of buildings benefitting from a combination of renewable heat in individual buildings, particularly heat pumps," p93.

It was with this background that the current PhD research set out to capture the reasons why heat pump technology is being promoted by the UK Government as an EU member state, why there is such optimism amid pessimistic UK trial results and how to move UK heat pump installation performance to match the best of European 'best practice'.

EST trial sponsor EDF-UK provided access to the EST trial data (2009-2010) and confidential trial reports. The initial plan was for a quantitative analysis of a large data set to provide detailed statistical evidence for a heat pump selection model. The STATA statistics package was employed to create two separate datasets of ground and air source heat pumps from the 85 excel files supplied, the resulting analysis providing annual performance for all the heat pumps. Whilst results were forthcoming, it became apparent that a range of design issues, primarily system boundary and monitoring protocol resulted in poorly conditioned results that invalidated modelled equations. Due to the initial design of the monitoring system, all heat pump efficiencies were expressed in the same measure of efficiency even where the actual installations were radically different. Within the trial there are a number of different heat pump types, heat pump combinations, mono and bivalent systems, space heating only and space heating with domestic hot water. The installations are set in buildings with wide ranging thermal efficiencies, hours of space heating operation, hot water consumption and secondary, unmetered, space heating. Whilst ostensibly the data provides a single measurement of heat pump "system efficiency" (SEFF), the nomenclature adopted by the EST, the significance of whole system design and system boundary becomes apparent.

The research therefore focused on the particular taxonomy of each installation and the monitoring protocol adopted at each site. Combining the quantitative spreadsheet data with qualitative information from the confidential trial reports results in observations primarily focused around the issues of system boundaries and of what is actually being measured and compared. It is also apparent that the quality of design, installation, commissioning and operation play significant roles in annual performance. An overview of the methodological approach provides triangulation between theoretical heat pump thermodynamics, the application of system type taxonomies and the training requirements for high quality system design and installation. The research outputs result in a review of field trial measurement techniques and their impact on trial results and training standards to identify heat pump installation potential, Figure 8-1. The research is a mixed methods approach where quantitative analysis provides numerical outputs that promote a qualitative response both at the trial population level, but also as grouped samples and individual case studies.

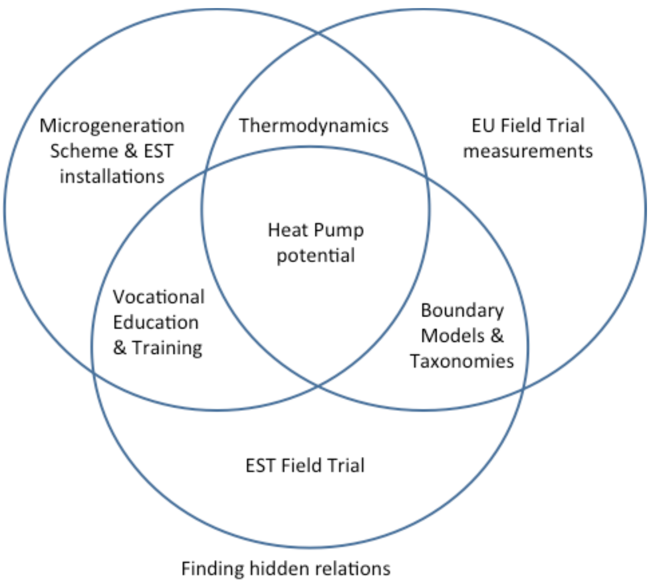


Figure 8—1 Thesis triangulation

The analysis of heat pump trials can provide valuable lessons in design and installation necessary for converting heat pump technology into a resource for tackling carbon dioxide emissions. It should be noted that these lessons are not limited to just this technology or to the UK. It is also possible that such an analysis will contain more general lessons for those attempting to introduce new energy technologies into demand sectors of the economy.

The research journey

This research is built around the results of the EST heat pump trials 2009 to 2010. These trials identified three significant outputs:

- the failure to report performance data to a boundary class based on the specifics of heat pump only performance and thus to enable comparison with other EU trials
- the failure to provide a significantly large and consistent sample such as monovalent space heating and domestic hot water. Without a consistent sampling approach combined with a transparent boundary analysis able to distinguish between heat pump, backup and circulator pumps, the trial data remains opaque
- that UK heat pump design and installation practice was generally poor in the period leading up to the 2009 - 2010 trials, thus raising uncertainties regarding UK vocational education and training (VET).

The journey to pursue a deeper understanding of these issues identified six specific objectives that derive from the research question in order to provide a contribution to knowledge, Figure 8-2:

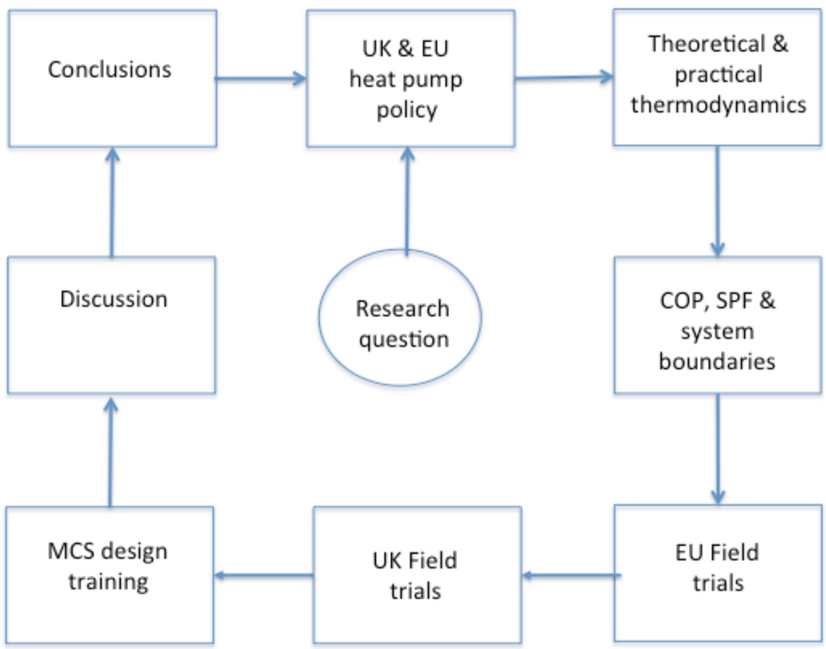


Figure 8—2 Thesis research journey

Research objectives

'What lessons can be learned from heat pump field trials and to what extent are field trial results being communicated in a consistent fashion?'

1. What is the UK central heating *status quo* and how does it relate to heat pump policy in a UK and EU context.
2. What does theoretical thermodynamics tell us about the efficiency of heat pumps and their practical limits?
3. What is the meaning of manufacturers' COP and its relation to SPF? What and where do we measure to identify as-installed efficiency for maximum analytical and practical benefit? What are the appropriate boundaries for pragmatic outcomes?
4. What do existing heat pump trials tell us? A comparison of real installations across Europe.
5. What does a deep analysis, both quantitative and qualitative, of EST raw data tell us?
6. What are the current MCS requirements for designing heat pumps, how do they reflect on the EST trial data and what is the appropriate form of VET?

Objective 1 Policy

UK central heating is dominated by the gas boiler and in particular the combi boiler. Combi boiler technology has evolved to provide a "system" package where all the component parts for a central heating system are packaged into a single boiler unit. Automatic control for output modulation and built-in diagnostics provide what could be described as a 'foolproof' solution. It is against this background that the heat pump is being promoted. In line with other EU member states, the UK is pursuing a policy of energy saving, carbon dioxide emissions reduction and the promotion of renewables. The EU directive on Renewable Energy Sources (RES, 2009) promotes the use of, *inter alia*, heat pumps with the Commissioners' decision of 2013 (EC, 2013) that minimum efficiencies be set at SCOPnet performance under EN 14825:2012*,* or in terms of measured performance at SEPEMO boundary SPF_{H2}.

The UK vehicle for renewable heat energy, the Renewable Heat Incentive (RHI), was originally planned for launching in July 2011 subject to the trial outcomes of the Renewable Heat Premium Payment scheme (RHPP). The RHPP, rather than providing a fixed kWh payment for renewable heat, provides an initial installation grant. The costs of the Feed in Tariff (FIT), the changes to the subsidy and resulting legal battles provide a scenario that, it is presumed, the UK Government does not wish to repeat, see for example, Daily Telegraph (Hollinshead, 2012). It is therefore assumed that RHPP results will be used to assess the long-term impact of renewable heat payments with the current launch date for the RHI household scheme planned for Spring 2014. All RHPP installations are by Microgeneration scheme (MCS) registered installers and, as such, will provide evidence of the general quality of installation, the current 'state of the art'.

Decarbonisation of the electricity grid, the primary issue when comparing emissions from heating between electricity and fossil fuels, whilst outside the scope of this research, will be dependent on a complex generating matrix of fossil fuels, nuclear and renewables. Allied to this is energy savings from retrofit, although the effect is indirect. Retrofit reduces the growth rate of demand, and makes it possible for a low carbon investment programme of a fixed size to catch up with demand more quickly. Calculations for renewable heat are based on EU-wide assessment of the ratio between total gross electricity production and the primary energy consumption for that electricity production, signified as η (eta), and based on an EU average compiled from Eurostat data. As the η ratio increases with decarbonisation, the minimum SPF for renewable heat will decrease in recognition of a lower benchmark requirement. Graphical analysis of this phenomenon is provided by Lowe (Lowe, 2007 p418) with comparisons between heat pumps, resistance heating and gas boilers, Figure 8-3.

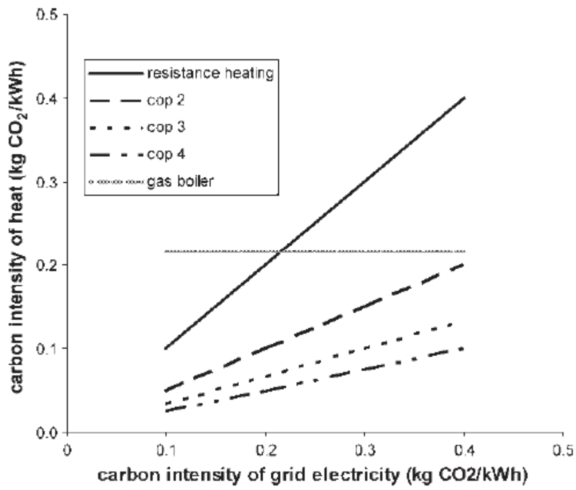


Figure 6 Carbon intensity of space and water heating: comparison of electric resistance heating, heat pumps with annual coefficients of performance (COPs) from 2 to 4, and a gas boiler with annual efficiency rate of 90%. Note that the electricity required to circulate hot water is ignored in all cases

Figure 8—3 Carbon intensity of the grid versus carbon intensity of heating appliances (Lowe, 2007)

Objective 2 Thermodynamics

The thermodynamics of the reversible heat engine would indicate that efficiencies could be much higher than those observed in practice. Carnot efficiency is based, among other things, on a theoretical temperature differential of almost zero ($\delta T \rightarrow 0$) across the system boundary whereas real heat pumps have a finite temperature difference (ΔT) across the evaporator and condenser heat exchangers where the rate of heat transfer is based on the log mean temperature difference between the heat exchanging fluids. All heat pumps in the EST trials are sub-critical vapour compression systems where assessment of COP is suited to pressure-enthalpy diagrams. For fixed speed compressors operating between the evaporator and condenser, a typical HFC refrigerant will reach about 60°C. Dropping the sink temperature from 55 to 35°C increases the log mean temperature difference and the rate of heat transfer driven by this temperature difference by a factor of four. To fulfil the same duty, low sink temperatures increase heat transfer for the same amount of fixed speed compressor 'work-in'. Further efficiency savings should be available for variable speed compressors and electronic expansion valves where the varying heat sink load can be matched to compressor power and super-heat controlled to minimise vapour and compressor overheating and its impact on polytropic compression. Unfortunately, the three heat pumps in the EST trials with both variable speed compressors and electronic expansion valves fail to show their potential for a variety of reasons including poor boundary definition, data logging failure and over-reliance on immersion heating.

For radiator systems in particular, which can operate across a wide range of mean water temperatures, weather compensation control will result in improved efficiency by varying sink temperatures to match heat losses provided that some form of feedback mechanism is fitted (such as a room thermostat) to ensure adequate response to thermal mass and internal heat gain impacts on room temperatures and thermal comfort.

For domestic hot water (DHW), de-superheating has the potential to raise the storage temperature and thus reduce the reliance on backup heaters for legionella control. Unfortunately, due to lack of appropriate monitoring, the two de-superheating units in the trial provide no evidence to quantify this impact.

Objective 3 COP, SPF and monitoring

The pursuit of real world performance requires an understanding of manufacturers' claims for coefficient of performance (COP) and its relation to seasonal performance factor (SPF). COP is a laboratory test where the heat pump is loaded to its maximum output at set source and sink temperatures, run for a set time period and the efficiency derived from the ratio of energy out to energy in. The test regime ensures maximum heat transfer during the process and inputs are based on compressor load and water flow resistance through just the heat exchangers, the evaporator and condenser. For air source heat pumps tested at low air temperatures, any defrost cycles are included into this calculation of COP. SPF, in contrast, is the annual energy efficiency for a system which will be operating across a range of loading and where all energy inputs may be included.

The thesis compares different approaches to measuring COP and the theoretical assessment of SPF based on COP values. A description is given of a BSRIA laboratory test for a manufacturer to EN 14511:2007. The same test is applied by BRE for a different unit but including an assessment of COP for DHW where the cylinder is heated from cold to hot. A description of the Passivhaus Institute testing method for a 'compact unit' is provided with outputs for both space heating and DHW at temperatures representing annual outdoor conditions ranging from (-7) to 20°C. DHW is tested for COP at both 'cold to hot' and, importantly, at 'reheat' conditions, a test more closely aligned to the current standard, EN 16147:2011.

The thesis also describes the pilot project experimental work carried out by the author in 2010 at the Barrett Green House, BRE, UK. Importantly, this work provides evidence which confirms that for stratified cylinders, those without a shunt pump to circulate water within the cylinder, there is little difference in COP between whole cylinder heating from cold and half cylinder re-heating following tapping since with typical cylinder design the primary coil is, in both cases, immersed in cold feed water and subject to the same temperature difference driving heat exchange. This pilot study indicates that for stratified cylinders any large draw off will result in heating from cold, therefore resulting in higher efficiencies and promoting the installation of cylinders that are controlled so that they are reheated only when their hot water content is at least half drawn-off.

In practice, only Passivhaus certificates provide COP values for both space heating and DHW as standard, although Passivhaus only provides this data for the compact or combined unit heat pump, comprising of a combination of heat pump, MVHR unit and cylinder, and originally developed by the Austrian company Drexel und Weiss for Passivhaus applications. Similar data is provided by the *Wärmepumpen-Testzentrum*, WPZ, Switzerland, based on testing at both EN 14511 and EN 16147 for air and ground source units. It is therefore possible to use COP test data to calculate a theoretical SPF for space heating and DHW based on the EN 15316 "bin method". A UK SAP analysis of a dwelling will provide an annual assessment of space heat and DHW loads in kWh. The varying space heat load and the DHW base-load can then be divided into temperature bins at the relevant COP for ambient temperature bands. Such a process is applied to air source heat pumps to provide seasonal COP or SCOP. However, each COP test value is for full load conditions and therefore different from that experienced when operating at less than full load, during real operating conditions, a situation which points to the obvious advantages of assessing SPF from field trial measurements including and excluding the loads from backup and circulation pump.

Perhaps most importantly, this chapter provides feedback on the practice of monitoring. The experimental work at the Barrett Green House identified the challenges in designing any form of monitoring system. The experience of making incorrect assumptions based on poorly understood mechanical design and/or installation monitoring protocol provide a wider resonance applicable to all field trials. It is imperative that the designers of trials understand the mechanics of each heat pump and its build specification in order to capture all inputs and outputs; it is clear that this is not the case. Equally, designers should not presume that installers of monitoring equipment will necessarily understand the objectives of the trial and thus the correct set-up of the monitoring system and finally, designers should double check results by resorting to analysis from 'first principles' as was demonstrated with Carnot efficiency based on reservoir temperatures.

Objective 4 Meta-analysis of EU trials

The research addresses system boundaries, both historic and current. Thirteen boundaries are identified from a review of trial literature ranging from the German *Jahresarbeitzahlen* (JAZ) and its sub-boundaries of JAZ 1, JAZ 2, JAZ 3 and JAZ 4, through to those promoted by the current SEPEMO programme, SPF_{H1}, SPF_{H2}, SPF_{H3} and SPF_{H4} (Zotl & Nordman, 2012). JAZ 1 and JAZ 2 are the subject of heat pump trials from Lahr in Germany (Auer & Schote, 2009). Only JAZ 2 is provided by Swiss FAWA trials (Erbe, et al, 2004), currently the largest database of heat pump performance. Fraunhofer trials (Russ, et al, 2010 and Miara, et al, 2011) focus on SPF_{H3} and Danish trials (Pederson & Jacobsen, 2011) on SPF_{H4}. The EST field trial (EST, 2010 and Dunbabbin & Wickins, 2012) provides results based on "system efficiency" (SEFF), a whole system boundary analysis that can prove to be non-transparent in terms of system components and heat pump only outputs. Efficiencies are derived from the combination of all inputs, both heat pump and backup, and with outputs from space heating and DHW hot water draw-off. Perhaps the greatest drawback to SEFF is the inclusion of DHW hot water draw-off within the system boundary since DHW cylinder losses are unaccounted for in the calculation of system output. SEFF, however, is the most representative boundary for measuring whole system efficiency, useful output/all inputs.

The European meta-analysis provides evidence of annual operation for over 600 heat pumps. Improved results are apparent from those published by EST where monitored data can be reassigned from SEFF to JAZ or SEPEMO boundaries. The mean for EST ground source heat pumps, irrespective of boundary, proves to be considerably inferior to all other European trials whilst the mean for air source heat pumps is closer although still lower. All trials indicate a wide range of performance indicating a sensitivity to context, however, the difficulty of manipulating trial averages to provide a single comparable performance unit across different boundary protocols, proved to be beyond this author (Gleeson & Lowe, 2013).

All boundary definitions have some ambiguity in terms of system components and their individual impacts. The importance of an "all boundaries" assessment of SPF provides the logic for creating "SPF_{H5}" (including DHW hot water draw off) and thus the full-cost boundary for the occupant. Only by measuring all inputs and outputs, in effect, from SPF_{H1} to SPF_{H5} can the impact of each load be assessed and diagnostics run to identify, for example, the load from the ground loop circulation pump, the extent of backup heating, the impact of distribution circulation pumps or the DHW hot water usage pattern of the household; all of which are secondary to heat pump performance although of great concern to the occupier since all contribute to the cost of operation. The important point raised in the analysis is that these different loads obey different physical and engineering laws and each therefore requires a different approach. A single, all-category metric, effectively muddles them up and makes it impossible to evaluate the impact of individual roles. Achieving such clarity exposes the conflict for manufacturers of trading off system integration, small footprint at low cost, with system transparency, where data logging and analysis functions are built into the unit thus making it possible to monitor without intrusive interventions within the unit, Figure 8-4. If manufacturers do not embed data capture systems into highly integrated systems, it will be impossible to do so subsequently. But with unintegrated systems, in which plenty of air and long lengths of pipework separate key components, it becomes possible for the researcher to retrospectively insert sensors between components, where the manufacturer has not done so.

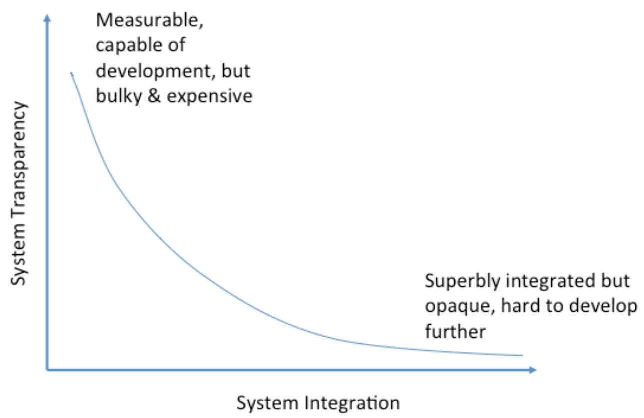


Figure 8—4 Conflict of integration versus transparency

Objective 5 Deep analysis of EST data

The confidential EST raw quantitative data from 2009 to 2010 and the qualitative reports from the EST trials were made available to the author (see acknowledgements). The trial data in excel files, provided at 5 minute time intervals for 85 heat pumps, is the subject of a deeper analysis originally premised on statistical sampling for 71 of those units. The initial process was to compile single data sets for both ground and air source heat pumps using the STATA statistical package. Each system's seasonal efficiency could then be analysed by applying boundary classifications to the individual monitoring protocols encountered. However, statistical analysis of data is dependent on the sample size and on data classifications that make sense. It is clear from the EST published report (EST, 2010) that the single classification of SEFF for ground and air source is unable to identify the variables associated with performance and therefore the data requires another approach.

Analysis by boundary, as with the European meta-analysis, results in some boundaries becoming redundant as systems fall into the common boundaries associated with SEPEMO. All SEFF values can be reclassified as either SPF_{H4}, systems either without DHW (space heating only) or where primary flow (heat into the cylinder) is separately monitored, or as SPF_{H5}, with DHW draw-off. Other SEPEMO boundaries are also evident but the scarcity of numbers in each classification makes any meaningful pronouncement somewhat dubious. An example is provided by Figure 8-5, where box plots for all 24 air source heat pumps provide what appear at first sight to be a reasonably logical relationship between means and percentiles for each boundary classification. However, only 4 units provide data for SPF_{H2}, 17% of the population. All 18 units for SPF_{H3} are estimated by subtraction of an unmeasured and therefore estimated mean circulation pump load. SPF_{H4} has only 7 units, 29% or less than one third of the population. SPFhps has 10 units and SPF_{H5} 12 units, however, both are non-SEPEMO boundaries and therefore not possible to compare to other large trial results. The graphical analysis does not include any reference to missing data. As an example, all boundary results for air source unit ID Code 473, ostensibly a poor performer, actually have 23% of the output energy data missing. With such small samples, there is a possibility that the missing data are not random, thus potentially biasing any statistics calculated from the remaining data.

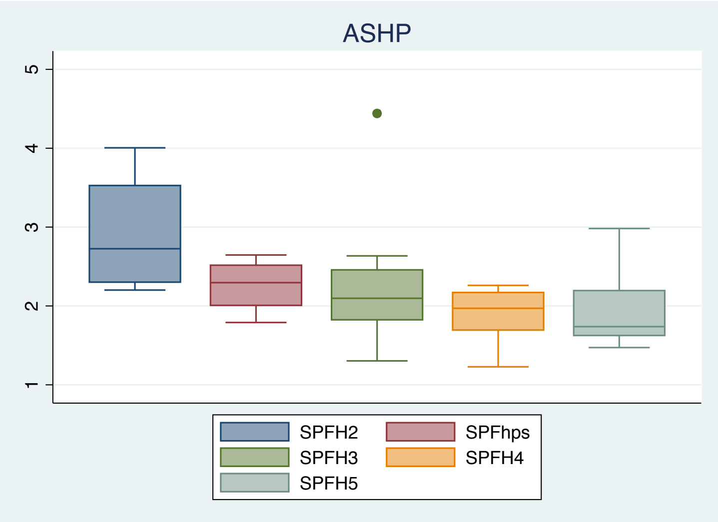


Figure 8—5 Air source heat pumps, total population

A similar story can be told of the ground source units, Figure 8-6. Here, for example, the mean for SPF_{H2} is raised by the presence of what is clearly a single outlier of 5.6 in a small data set of 10 out of 51 units. It is worth noting that this outlier at SPF_{H2} is missing 24% of the heat pump input energy and that it appears likely that this is the reason for its high efficiency. Without the outlier, the mean for SPF_{H2} drops from 2.9 to 2.6. Estimating SPF_{H3}, in the same manner as with air source heat pumps, provides some means that are lower than those in SPF_{H4} and SPF_{H5} and has therefore been discarded since this is illogical. SPF_{H4} has a smaller variance but lower mean than SPF_{H2}, implying that this sample as a whole, produces better results than SPF_{H2} - "implying", of course, whilst remembering the impact of the SPF_{H4} outlier. A statistical approach provides some lovely illustrations, however, conclusions from a solely statistical approach are unlikely to be robust in the face of poorly understood features in datasets and measurement protocols.

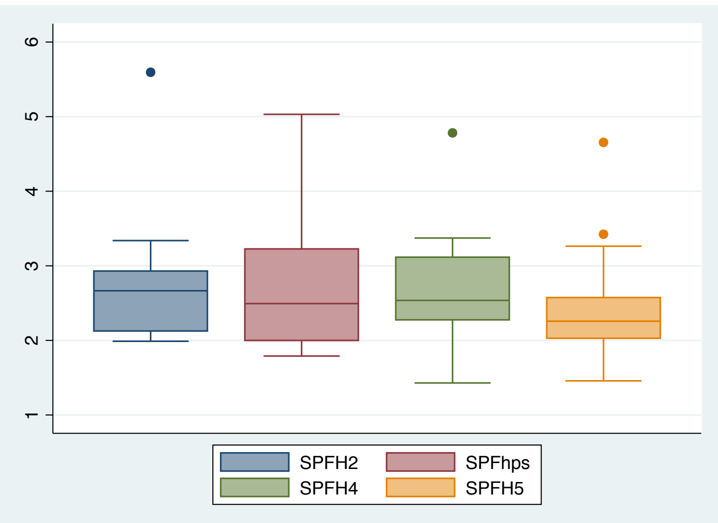


Figure 8—6 Ground source heat pumps total population

A taxonomy of system morphologies has therefore been developed, a classification process to group each system by source, mono or bivalent, space heating only or space heating with DHW. Each class of system can then be assigned a boundary depending on the monitoring technique applied during the trial. This exercise establishes that the trial design, by including too many system typologies and by trying to answer too many questions with inadequate metering, suffers from having a very small quantifiable sample of heat pump-only central heating systems, thus risking the core objective of the trial.

The results of the taxonomical analysis provide evidence that the trial was not primarily designed to assess heat pumps as an alternative to monovalent gas boiler central heating with both space heating and domestic hot water, as is evident in Figure 8-7.

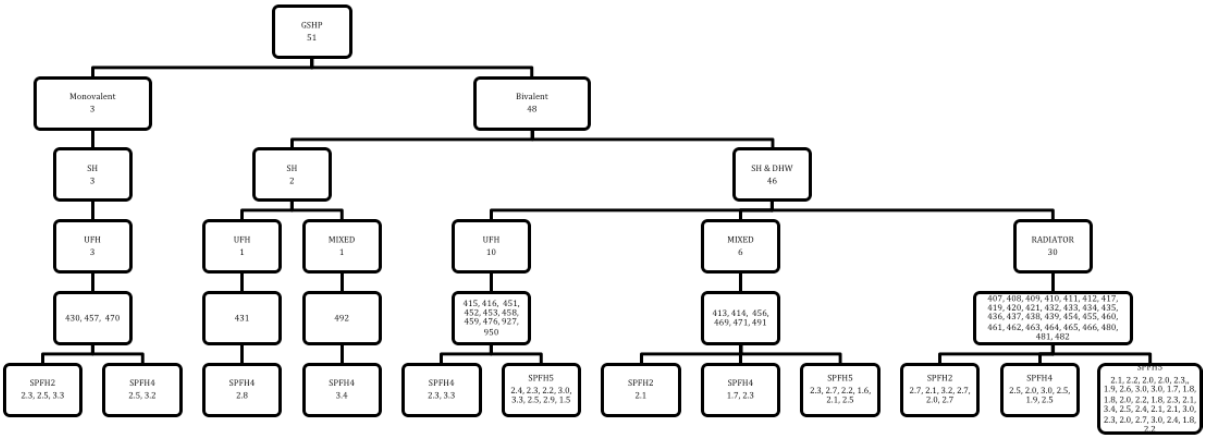


Figure 8—7 GSHP taxonomical output (from Figure 6-4, p166)

The ground source heat pump taxonomy provides no monovalent space heating and domestic hot water units. Any analysis of bivalent heat pump performance must rely on a monitoring system capable of distinguishing between the heat pump and the backup. Only seven out of fifty-one units provide heat pump only efficiency at the SPF_{H2} boundary. All the others include a range of backup, circulation pump and hot water draw-off rather than heat into the cylinder. The taxonomical analysis of air source heat pumps identifies SPF_{H2} in only two out of twenty four units. A satisfactory statistical analysis of heat pump efficiency is thwarted by the initial trial sample selection and monitoring protocol.

The alternative to a statistical approach is to treat each system as a case study. Flyvberg (2006), for example, supports the use of case study in the social sciences since:

> "[C]ase study produces the type of context dependent knowledge that research on learning shows to be necessary to allow people to develop from rule-based beginners to virtuoso experts," p221.

By combining building services engineering design techniques, with the output of a heat pump-focused literature review and the qualitative information available from the confidential trial reports, such as photographs and written descriptions, a set of design and operation questions is formulated. The fundamental difficulty of associating poor performance with specific features of system design (with a view to changing or deselecting those features) is clearly the lack of monitoring data at the appropriate boundaries. However, analysis of these questions provides ample evidence of poor system design including the low correlation between building heat loss and heat pump power, between heat pump power and ground loop length, between sink temperature and SPF. There is no consistency in the application of buffer vessels where some feed the entire system, some the space heating only and, remarkably, some the DHW cylinder circuit.

Individual systems are interrogated to provide evidence of cycling, continuous and intermittent operation along with energy tariff and system control. The results indicate a pervasive failure in design and in controls specification, not always the fault of the heat pump installer. Controls remain a black box for many occupiers and, it is suspected, for installers. Manufacturers' instructions for controlling the heat pump system range from the technically sophisticated, but written in terms few would either understand or be bothered to work through (one manufacturer provides 10 pages of detailed instructions for setting the controls), through to little or no advice on system operation and matching controls to tariff. An example of the more sophisticated issues which impact on SPF include fine-tuning weather compensation curves to building thermal mass and thermal half-life response, a responsibility which, if done at all, inevitably falls on the dwelling occupant.

Any addition to heat pump power requirements for DHW load should be dependent on a range of issues including the building thermal mass but also the specifics of the hot water system including cylinder size and performance characteristics, DHW usage and re-heat pattern. These issues are not evident in manufacturers' literature or the trial published reports, either public or confidential.

The analysis by taxonomy shows that there is an insufficient sample of heat pump only systems measured at the appropriate boundary to provide definitive evidence of heat pump performance. The combination of the trial sample selection and monitoring implementation is therefore flawed. Whatever information is extracted from the data, suffers from the inadequacies of the measuring and reporting methodology.

Objective 6 MCS Accredited installer requirements

The design issues identified from the EST data are subjected to analysis by applying the Microgeneration scheme “installer standards” and “reference materials”. For registration as an accredited installer with the MCS scheme, a company must provide evidence of quality assurance and appropriate training. MCS provide comprehensive documentation, supported by webcasts, to assist the designer through heat loss calculations and the assessment of annual energy demand. It is apparent that the designer must be able to apply the design criteria for selection of heat pump (MIS 3005, 2013) in whatever format provided by manufacturers whether graphical or tabulated. The documentation provides guidance for radiator and cylinder sizing but does not mention buffer vessels, tariffs or controls. For ground source units the documents provide sufficient detail to size the ground loop length but no additional webcast support for those with no experience in calculating pipe sizes, flow rates, pressure drops and pump selection. Technically, most of the documentation and the supporting webcasts appears to be excellent (although not tested by the author) and continues to evolve.

MCS documentation may be compared with that of the European Heat Pump Association (EHPA) registration scheme, EUCERT. The MCS approach is utilitarian, its aim is to provide detailed guidance on designing for heat pumps, whereas, it is apparent from the training manual (available from ConstructionSkills, UK), that EUCERT is based on a more educational approach where heat pump designer/installers are expected to have *berufliche Handlungsfähigkeit* (Brockman et al, 2009), described as a multidimensional occupational capacity which integrates all knowledge, practical wisdom and understanding needed to practice an occupation.

The training manual provides information on marketing and financial analysis, energy efficiency, thermal comfort and carbon reduction as an introduction to the heat pump technically specific data that includes both thermodynamics and system design. The self-assessment exercises demand a high level of numerical manipulation based on a thorough understanding of themes that range from net present value to heat pump/heat loss power curves. Interestingly, although ground loop design is covered in the EHPA documentation, it is at a much less comprehensive level than in the MCS documents. The EHPA is currently developing ground loop design criteria and training through the Geotrainet project and, presumably, will publish education and training documents when this project is complete.

Applying MCS micro-installation standard MIS3005 to a design indicates that the designer must have a thorough grasp of engineering principles along with a high level of "*functional skills*" (OFQUAL, 2010) in mathematics and information and communication technology (ICT). Prerequisites for what is described by providers as "MCS short course training", generally between 3 and 5 days, indicate to this author, that those on the "*experienced worker route*"[43] (that is, existing installers with no formal qualifications) and even those with NVQ Level 2 and Level 3 plumbing qualifications may struggle with "*synthesis and evaluation",* to invoke Bloom's taxonomy of learning (Bloom, 1954).

The impact of MCS design requirements should be evident in the performance of heat pumps installed by registered companies. All installations under the Renewable Heat Premium Payment scheme (RHPP) are by registered installers. There are plans for up to 700 metered installations and, it is hoped, the monitored performance will be made available for analysis by independent researchers. Currently (February 2014) provisional results from the RHPP are being treated as confidential.

A comparison between the Fraunhofer trials, identified as having the highest SPFs in the meta-analysis, and the RHPP results may shed some light on the impact of vocational education and training between the MCS task-based skills approach in the UK and the occupational capacity approach associated with the Continental model, although no direct link to EUCERT is evident. The demands for heat pump design, as spelled out in the MCS documentation require a high standard of thermal literacy and engineering education. The low level prerequisites from UK training providers indicate that successful completion of courses does not necessarily imply understanding of the design process - one of iterations, comparisons and decision making.

Conclusions

The analysis of heat pump trials can provide valuable lessons in design and installation necessary for converting this technology into a resource for tackling carbon dioxide emissions. It should be noted that these lessons are not limited to the UK. It is also possible that such an analysis will contain more general lessons for those attempting to introduce new energy technologies into demand sectors of the economy. The research has identified the need for care with definitions as the basis for coherent and useful comparisons; the trade off between product integration and measurability and comprehensibility; the risk of assuming that existing supply

[43] "Experienced worker group" MCS Heat Pump Working Group minutes, 13/12/2012, p7.

chains can cope with the challenges of new technologies without significant up-skilling. One could argue of the UK, and the UK is by no means unique, that there exists a general lack of technical capacity in supply chains, academia and government, to install, measure, analyse and interpret the performance of new technologies. This lack of technical capacity has been persistently confirmed from Rosenfeld in 1999 to Oreszczyn and Lowe in 2009:

"For me, the most interesting outcome was not the official one, which was that an alert, motivated design team could save 50% of the energy with a reasonable payback time, but was how hard it was to find any competent design team and any competent "third party" to do the measurement and verification," (Rosenfeld, 1999 p69)

"Buildings and energy research has for many years more closely resembled a cottage industry than a mature academic sector, with few, large-scale or rigorous investigations, limited high-quality science and weakness in formal structures to promote and sustain a common research culture…. The lack of data on energy performance of new and existing housing is symptomatic of a long-term lack of UK research into low energy buildings," (Oreszczyn & Lowe, 2009 p112).

Heat pumps and other emerging, underdeveloped technologies, require installers to understand the underlying thermodynamics, "what makes it low energy" and the mechanics of the system, "what makes it work". With the appropriate knowledge, skills and competence it is possible to provide transparency of measurement, to quantify all energy inputs and outputs and to provide feedback to policy makers, contractors and the end user. This process is underway; MCS documentation and competency requirements are dynamic and evolving to reflect a growing understanding of how to achieve low energy goals. The thesis attempts to show that by focussing on the issues that have arisen with what may be the most complex heating technology yet introduced to the housing sector, it may be possible to anticipate and forestall future failures.

The research has shown that complex mechanical systems operating at small temperature gradients require a level of technical design ability that is more closely associated with formal engineering qualifications. The imposition of such a demand on the UK domestic heating industry would require the development of a deeper educational input into mainstream VET along with control mechanisms such as individual, as opposed to company, registration of designers and installers, similar to that for gas installers. VET providers, both private and public, also need to be rigorously assessed in terms of staff technical ability and training provisions. There would be significant social impacts, not the least of which would be the increase in employment costs of higher-qualified operatives. Such a demand and its consequences resonate with the continental VET models referred to in the text where, in Germany for instance, VET content is negotiated between the Federal Government, social partners (employers and trade unions) and educationalists; in Britain it is solely employer-led.

Manufacturers have the opportunity to offer design services as part of their sales package, formally supported with quality assurance of the design. Such an approach requires a full assessment of the building loads for heat pump sizing, the matching of emitters to heat loss parameter along with control settings for optimised performance; a design package requiring an equally competent installer. Ground loop and borehole contractors could offer a similar service.

Self reflection

The research has attempted to take a mixed methods approach to the problem of assessing the output of field trial measurements. The key to the analysis is the taxonomic comparison of the European trial outputs with that of the EST. Data uncertainty, whilst recognised, has not been formally addressed either in terms of data cleansing or as core requirement for a more meaningful multivariate statistical approach to the EST data that attempts to define the key independent variables and their relative impact on the dependent variables of instantaneous COP and SPF, the original aims of the research. This does not, however, exclude the possibility that useful information can still be extracted by developing multiple regressions, machine learning and multilevel statistical modelling; a multilevel analysis could well provide significant output from the EST data and useful guidance for system design and control.

As with many in-depth research procedures parallel themes have come to light, the importance of which were not obvious to the author at planning stage. The author has become aware of research disciplines focused on the “social construction of technical systems” (Bijker, et al, 1989) and the synergies of “technological transitions and system innovations” (Geels, 2008). The impact of renewable technologies, non-fossil fuel societies, implies deep social change in the supply chain and wider stakeholders. The debate on ‘competence’ in vocational education and training places heat pump technology, its design, installation and operation, within this paradigm. The thesis has attempted to navigate through multiple factors, to map their influences and direction on installation performance. The breadth of the research suggests a somewhat unsatisfactory treatment of each component part although still providing a conjunction or alignment of critical issues affecting field trial performance.

The contribution to knowledge

Proof of heat pump sensitivity to context

The research has shown that heat pump central heating installations are sensitive to design and installation in ways that competitors, gas, oil and electric resistance heating, are not. There are no published results for large scale oil-fired central heating trials, however, the EST gas condensing boiler trials (EST 2009), reports efficiencies based on the inclusion of domestic hot water:

> "[A] more valid comparison of regular and combination boiler annual efficiency may be 80.3% compared with 82.5%."

The report provides standard deviations for regular boilers of 2.5% and combination boilers of 4%. Whilst the EST boiler trials cannot be directly compared to the heat pump trials, since the boiler trial allowance for domestic hot water losses are estimated and do not include electrical loads, the heat pump trials provide SPF_{H5} means for ground source and air source of 230% (2.3) and 180% (1.8) respectively, both with 50% (0.5) standard deviation. Applying these standard deviations to 95% of the population provides a range of difference of 10% for regular boilers, 16% for combis but 200% for heat pumps, Table 8-1.

Appliance type	Standard deviation	95%	68%	mean	68%	95%	Range difference
Regular	2.5	75.3	77.8	80.3	82.8	85.3	10
Combi	4	74.5	78.5	82.5	86.5	90.5	16
GSHP	50	130	180	230	280	330	200
ASHP	50	80	130	180	230	280	200

Table 8—1 Gas boilers and heat pumps, comparison of sensitivity

In terms of theory, heat pumps, in contrast with gas boilers, are sensitive to context because of the number of additional degrees of freedom. A simple analogy is the difference between a train on a straight track and a ship at sea. The train can only move forward or backward, it has 2 degrees of freedom, whereas a ship is subject to an array of forces: "[which] may arise from pitching, rolling, heaving, surging, yawing or swaying or a combination of any two or more"[44]; it has 6 degrees of freedom. Predicting the fuel efficiency of a ship or its position in space and time is therefore more complex than that of a train. A gas boiler has just one heat transfer process, between the fuel and the heat exchanger containing central heating water, which we may describe as 1 degree of freedom. Within the boundary of the heat pump system SPF_{H2}, an ASHP has 2 energy sources, ambient heat and electricity. Heat transfer occurs from the source to

[44] http://www.pomorci.com/Zanimljivosti/Ship%27s%20movements%20at%20sea.pdf

evaporator, compressor heat to the refrigerant fluid and condenser heat to the central heating water, 3 degrees of freedom. A GSHP has 4 degrees of freedom, the same as the air source but with the additional heat transfer from the ground to the ground loop. Each process of heat transfer across a temperature gradient, combined with friction and heat losses, generates entropy. Even a perfectly designed heat pump system must operate within these constraints. Each poor design decision such as a mismatch between heat power and building heat loss, an undersized ground loop, high pumping loads, high temperature sink, undersized emitters, high electrical resistance load, will further downgrade the SPF. At the most fundamental level, the sensitivity is based on the temperature gradient across the heat exchanger. The temperature difference across heat pump heat exchangers is at most 10s of degrees whereas that of boilers is 100s of degrees.

The central role of taxonomy

The key to understanding heat pump performance is in understanding heat pump components, system morphology and measurement at appropriate boundaries. The development of a taxonomy has benefits at the design stage, in selecting potential installations and in assessing critical data outputs. Whilst SPF_{H2}, the RES 2009 boundary, provides the heat pump efficiency, the impact of any backup heaters and the circulation pump are lost. SPF_{H2} can classify a system as renewable even when there is a high resistance load and may therefore lead to an over-estimation of carbon dioxide savings from heat pump installations. SPF_{H3} alone cannot distinguish between heat pump and backup. SPF_{H4}, whilst a useful assessment of whole system operation, also suffers from the inability to identify backup and circulation pump impacts. SPF_{H5}, the non-SEPEMO but useful boundary that includes domestic hot water draw-off, provides the occupier with their real world efficiency but one which is dependent on the specifics of their pattern of hot water use and the cylinder specification. The EST assignment of SEFF in the original trial analysis is of little use in analysing trial impacts since it lumps all systems into a single classification whether or not they have backup heating, supply space heat only or supply both space heating and domestic hot water; however, the same argument applies to SEPEMO's SPF_{H4}. For full transparency of operation, energy flows at all boundaries need to be measured, a taxonomical approach helps to ensures that outcome.

DECC and EST have changed their trial boundary analysis to reflect the above arguments. Although not claiming to be the sole driving force behind this decision, the author did present the above analysis to trial sponsors EDF as far back as November 2011.

Vocational education and training

It is possible to achieve what is classified as *renewable heat* across all heat pump installations if appropriate attention is focused on the design, installation and control of heat pump systems. However, even in comparable trials with a high mean SPF, there still exists a wide range of performance where, due to the sensitivities identified, individual systems have low efficiencies and therefore high running costs and carbon dioxide emissions. The reliance on heat pumps to reduce fuel poverty will result in greater fuel poverty for some occupiers unless the requirements for appropriate design and installation are met. Requirements for high performance are premised on appropriate vocational education and training (VET) with a high educational content. The UK approach, which includes those with no formal training, known as "*the existing worker route*", necessitates a rote-learning approach premised on meeting a tick-box assessment of competence. Such an approach is fraught with the dangers of "*garbage in - garbage out*"[45], a warning provided in the MCS webcasts on heat loss calculations and where design calculations carried out after training, may have no theoretical foundations underlying their meaning. Evidence of the quality of current MCS accreditation standards will hopefully appear when the RHPP results are eventually published. There are, as yet, no legally binding demands on installers that systems must be rectified when they do not meet SPF targets and where annual running costs are significantly greater than those specified. For such a ruling, the difficulty would be in proving that the performance gap lay in the system design rather than in occupant use of the system. It would arguably take as much skill to decide this as to install the heat pump properly; we are back with Rosenfeld's lament on the scarcity of human resources.

Heat Pumps and Domestic Hot Water

The pursuit of SPF for DHW production finds the literature distinguishing between heating from cold and re-heating after tapping. Field trial results from the Barratt Green House experiments indicate that the tapping test for both EN 255-3:1997 and EN 16147:2011 are applicable only to cylinders with shunt pumps where there is constant circulation of water within the cylinder. Both require the draw-off of water until the cylinder temperature drops to 40°C. For stratified cylinders, the dominant UK design, at this point the primary coil will generally be immersed in cold feed water and the re-heat will exhibit the same COP features as heating from cold. As a corollary of this observation, installing a large stratified cylinder that can be heated infrequently optimises the efficiency of hot water production.

[45] http://www.microgenerationcertification.org/mcs-standards/reference-materials/heat-pump-reference-materials [accessed 23 September 2013]

Culture shift

The cultures of the UK plumbing profession, and no doubt other countries, along with the training systems that support it (including concepts such as VET), have co-evolved with the technology of the dominant heating source, in the UK the gas boiler, over a period of 50 years. At the beginning of this period, the technology was much simpler than it is now. The introduction of heat pumps is taking a different approach, with a technology that is inherently complex from the outset, in the context of an energy policy that requires the more or less complete replacement of gas boilers by heat pumps in around three decades.

Future research

UK heat pump performance

The research is but a snapshot in the historical development of the heat pump as an emerging technology in the UK and elsewhere. Whilst developing the research thesis, the legislative requirements for the Directive on the promotion of the use of energy from Renewable Energy Sources (RES, 2009) have been clarified. The European Commission's decision of March 2013 has set SPF_{H2} as the monitored boundary for assessment of renewable heat energy and SEPEMO have provided the trial methodology. At the date of writing, some 700 heat pumps are being monitored for performance by DECC under the RHPP, all of which are MCS accredited. The cost of the RHI to the Treasury will depend on the minimum SPF_{H2} value ascribed to renewable heat. For the UK, the Commission decision has been interpreted by DECC in their EST Phase 2 report (Dunbabbin, et al, 2013) as:

"The European Commission states that the minimum level of SCOP for a heat pump to be considered renewable is 2.5. The same document indicates that the system boundaries for this calculation are those of SPF_{H2}."

It is important to note that the values of SPF_{H2} that actually should be used are based on the UK climate zone classification as "average" and "warmer" rather than, as DECC have assumed, as "cold". Therefore SPF_{H2} should be 2.6 and 2.7 for air and 3.5 for ground source heat pumps, not a single value of 2.5 for all heat pumps. This author was invited to review both the DECC (2013) and the EST (2013) Phase 2 reports. At the time of reviewing the DECC Phase 2 report the author missed highlighting this anomaly due to focusing on the heat pump performance aspects.

The collection of renewable data based on estimated SCOPnet would alleviate installers from the need for monitoring systems and save the costs associated with data logging, data collection and data analysis. RES outputs could be supplied by reference to heat pump sales rather than as-installed SPF_{H2}. However, this option should only be considered when there is definitive evidence of improved design and operation.

The performance gap and VET

The issues surrounding heat pump performance and design/installation quality are reflected in the parallel debates around "*nearly-zero energy buildings*" (EPBD, 2010 paragraph 17, L153/15), the UK zero carbon homes standard for 2016 and the UK Green Deal. Although the performance gap has been suggested by previous authors (Markus & Morris (1980), Lowe & Bell (1998)) , work by Leeds Metropolitan University in particular, has begun to quantify what has become widely known as "*the performance gap*", the gap between design and performance in both new build and retrofit. DECC have recognised the gap and have provided "in-use factors", to apply to retrofitted energy saving interventions through the Green Deal (DECC, 2012 p9). The discussion provides six reasons for the performance gap, with "imperfect installations" as just one of those reasons. It is clear from such studies as the Stamford Brook project (Wingfield, et al, 2008) that responding to imperfect installations with clearly stated energy objectives supported by, for example, tool-box talks and feedback from on-site testing can help to reduce the performance gap; the performance gap is by no means fixed. The analysis of EST trial results emphasises the role of DECC's "imperfect installations" and justifies further research on UK vocational education and training and the role of the Continental VET concept of "*occupational capacity*" especially in respect to the European Qualifications Framework. Perhaps all construction workers, certainly those who work on the building envelope and energy services, require an evolving "thermal literacy" (Gleeson & Clarke, 2013) among their range of skills.

The analysis of heat pump trials can provide valuable lessons in design and installation necessary for converting *context sensitive* technologies into a resource for tackling carbon dioxide emissions. It should be noted that these lessons are not limited to the UK. It is also possible that such an analysis will contain more general lessons for those attempting to introduce new energy technologies into demand sectors of any economy. Examples abound: the need for care with definitions as the basis for coherent and useful comparisons; the trade off between product integration, measurability and comprehensibility; the risk of assuming that existing supply chains can cope with the challenges of new technologies without significant up-skilling; the general lack of technical capacity in supply chains, academia and government, to install, measure, analyse and interpret the performance of new technologies.

This list, whilst not exhaustive, is not intended to promote a pessimistic outlook. The point is that by focussing on the issues that have arisen with what may be the most complex heating technology yet introduced to the housing sector, it may be possible to anticipate and forestall future failures.

Chapter 9 References and Appendices

References

Actionenergy (undated) *Energy Consumption* [online] http://www.actonenergy.org.uk/energy-consumption [accessed 14 November 2014]

Afjei, T. (2002) *Seasonal performance calculation for residential heat pumps with combined space heating and domestic hot water production.* [online] http://www.fhnw.ch/habg/iebau/dokumente-1/afue/.aeudetechnik/wp_ww_jng_sb.pdf [accessed 18 September 2011]

ASD (2013) *Simplified Technical English*. Issue 6. [online] http://www.asd-ste100.org/ [accessed 5 September 2013]

Auer, F., Schote, H. (2009) *Schlussbericht Zweijähriger Feldtest Elektro – Wärmepumpen am Oberrhein: Nicht jede Wärmepumpe trägt zum Klimaschutz bei Erdreich-Wärmepumpen mit positiver Ökobilanz Kritische Bewertung von Luft-Wärmepumpen.*- Final Report Two-year field test electric - heat pumps on the Upper Rhine: Not every pump contributes to climate change in ground-heat pumps with positive ecological balance Critical evaluation of air source heat pumps. [online] http://www.agenda-energie-lahr.de/WP_Jahresbericht2006-08.html [accessed 30 September 2012]

Barratt Developments (undated), *Barratt Green House* [online] http://www.bre.co.uk/filelibrary/Innovation_Park/Barratts_Green_HouseBrochure.pdf [accessed 21 November 2013]

Baumgartner, T., Gabathuler, H. R., Szokody, G. (1993) *Wärmepumpen. Planung, Bau und Betrieb von Elektrowärmepumpenanlagen.* RAVEL im Wärmesektor. Heft 3. [online] http://www.energie.ch/phocadownload/356D.pdf [accessed 30 September 2012]

Baxi. (2006) *Operation, Installation & Maintenance Instructions: Installation of ground array (Slinky)* [online] http://www.baxi.co.uk/docs/Baxi_Geoflo_Ground_Array_%28Slinky%29_Installation_Instructions.pdf [accessed 5 September 2013]

BERR, (2008) *MCS Product Certification Scheme for heat pumps, MCS 007* [online] http://www.greenbooklive.com/filelibrary/MCS_007_-_Iss_1_2_Product_certification_scheme_requirements_-_Heat_Pumps_250208.pdf [accessed 18 September 2011]

BHF Unlimited (online) http://bhfunlimited.co.uk/Underfloor-Heating/underfloor-heating-Pipe?product_id=606&CAPCID=23304006860&cadevice=c&gclid=CMqZt9Tb3LgCFfMdtAod_EoAWw&CA_6C15C=1927450210 [accessed 5 September 2013]

Bidstrup, N., Seymour, D. (undated) *High efficiency circulators for domestic central heating systems* [online] http://www.bpma.org.uk/filemanager_net/files/id83_bidstrup_final1.pdf [accessed 5 September 2013]

Bijker, W, E., Hughes, T. P., Pinch, T. (1898) *The Social Construction of Technological Systems - New Directions in the Sociology and History of Technology*. The MIT Press

Blomberg, T. (1995) *Heat conduction in two and three dimensions: Computer modelling of building physics application.* Report TVBH-1008, ISRN LUTVDG/TVBH--96/1008--SE/(1-188), ISBN 91-88722-05-8. Department of Building Physics, Lund University, Sweden. [online] http://www.buildingphysics.com/manuals/avh_TB.pdf [accessed 6 September 2013]

Bloom, B. (1954) *Taxonomy of educational objectives*. Longmans. See for example: https://en.wikipedia.org/wiki/Bloom%27s_Taxonomy#cite_ref-2 [accessed 6 September 2013]

Boait P, J., Fan D., Stafford A. (2011) *Performance and control of domestic ground source heat pumps in retrofit installations.* Energy and Buildings 43 (2011) 1968-1976.

Boait P.J., Dixon D., Fan D., Stafford A. 2012. *Production efficiency of hot water for domestic use*. Energy and Buildings 54 (2012) 160-168

BRE, (2005) *Energy Use in Homes, A series of reports on domestic energy use in England. Space and Water Heating*. [online] http://www.bre.co.uk/filelibrary/pdf/rpts/Space_and_WaterHeating.pdf [accessed 18 September 2011]

BRE, (2007)[a]. *Mitsubishi PUHZ-W90VHA air to water heat pump tests. Report number 237184*. Edited edition [online] www.e-si.com/downloads/BRE%20Test%20Report%20Mitsi_W90.pdf [accessed 6 September 2013]

BRE, (2007)[b]. *Mitsubishi PUHZ-W90VHA air to water heat pump tests. Report number 237184*. Full report available in hard copy from Mitsubishi.

Brockman, M., Clarke, L., Winch C. (2009) *Competence and competency in the EQF and in European VET systems*. Journal of European Industrial Training Vol. 33 No. 8/9, 2009 pp787-799 Emerald Group Publishing Limited

Bruderer, H., Hohl H. (2008) *Effects of electronic expansion valves on heat pump performance*. [online] http://www.satagthermotechnik.ch/etc/medialib/internet_satag/Publikationen.Par.32523.File.File.tmp/IEA_hpc2008_manuscript.pdf [accessed 4 September 2013]

BS 5449:1990, *Specification for forced circulation hot water central heating systems for domestic premises.* BSI

BS 6700: 2006+A1:2009 *Specification for design, installation, testing and maintenance of services supplying water for domestic use within buildings and their curtilages.* BSI

BS EN 255-3:1997. *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors. Heating mode Part 3. Testing and requirements for sanitary hot water units*. BSI

BS EN 442-3:2003, *Specification for radiators and convectors. Evaluation of conformity.* BSI

BS EN 677:1988. *Gas-fired central heating boilers. Specific requirements for condensing boilers with a nominal heat input not exceeding 70 kW.* BSI

BS EN 806:2000. *Specifications for installations inside buildings conveying water for human consumption*. BSI

BS EN 1264-2:2008 + A1:2012, *Water based surface embedded heating and cooling systems. Floor heating: Prove methods for the determination of the thermal output using calculation and test methods.* BSI

BS EN 12309-1:2000 *Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW.* BSI

BS EN 12831:2003. *Heating systems in buildings - Method for calculation of the design heat load.* BSI

BS EN 14511:2007. *Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling: - Part 1: Terms and definitions - Part 2: Test conditions - Part 3: Test methods - Part 4: Requirements.* BSI

BS EN 14825:2012 *Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling. Testing and rating at part load conditions and calculation of seasonal performance*. BSI

BS EN 15316-4-2:2008, *Heating systems in buildings — Method for calculation of system energy requirements and system efficiencies — Part 4-2: Space heating generation systems, heat pump systems*. BSI

BS EN 15450:2007, *Heating systems in buildings — Design of heat pump heating systems.* BSI

BS EN 15502-2-1:2012. *Gas-fired central heating boilers. Specific standard for type C appliances and type B2, B3 and B5 appliances of a nominal heat input not exceeding 1000 kW.* BSI

BS EN 16147:2011*, Heat pumps with electrically driven compressors — Testing and requirements for marking of domestic hot water units*. BSI

BSRIA, (2009). Report 53448/2. *Test on air to water heat pump. Carried out for Eco Tec Heat Pumps. Compiled by Tom Garrigan*. [online] http://www.ecotec-heatpumps.com/About/documents/BSRIA_final_report_53448-2.pdf [accessed 6 September 2013]

Calorex Installation/technical manual operating instructions 3500dt 5000dt geothermal heat pump. Available from the manufacturer

Carbon Trust. (2007) *Micro-CHP Accelerator, Interim report*. [online] http://uspace.shef.ac.uk/docs/DOC-32821 [accessed 5 September 2013]

Cengel, Y. A., Boles, M. A. (1994). *Thermodynamics, an engineering approach*. 2nd Edition. McGraw-Hill

CIBSE Domestic Building Services Panel (2011) *Domestic Heating Design Guide* (DHDG)

CIBSE Domestic Building Services Panel. (2012) *Underfloor Heating: Design & Installation*

CIBSE. (2006) *Environmental design, CIBSE Guide A*

Clarke, A., Grant, N. (2010) [online] *The importance of hot water system design in the Passivhaus* http://www.elementalsolutions.co.uk/downloads/Dresden%20paper%20final%201.pdf [accessed 18 September 2011]

CLG, (2010) *English Housing Survey Annex 3 2010* [online] http://www.communities.gov.uk/documents/statistics/pdf/1479789.pdf [accessed 18 September 2011]

CLG, (2010) *English Housing Survey Headline Report 2008-09*. [online] http://www.communities.gov.uk/documents/statistics/pdf/1479789.pdf [accessed 18 September 2011]

ClimaCheck Performance Analyser. [online] http://www.climacheck.com/ [accessed 6 September 2013]

Colbourne, D. (2010) *Review of performance of electric heat pumps*. Available from the author at Re-phridge, PO Box 4745, Stratford-upon-Avon, Warwickshire, CV37 1FE, United Kingdom. d.colbourne@re-phridge.co.uk

Committee on Climate Change (2010) *The Fourth Carbon Budget: Reducing emissions through the 2020s* [online] http://www.theccc.org.uk/publication/the-fourth-carbon-budget-reducing-emissions-through-the-2020s-2/ [accessed 5 September 2013]

ConstructionSkills. (2010) *Sector Skills Assessment for the Construction Sector 2010*. [online] http://www.cskills.org/uploads/ssaukreport2010_tcm17-26611.pdf [accessed 5 September 2013]

Coolpack software. [online] http://en.ipu.dk/Indhold/refrigeration-and-energy-technology/coolpack.aspx [accessed 4 September 2013]

Daikin Altherma (undated) *Technical Data, EEDEN09-720* [online] http://server.mmc.bg/MMCpublic/CD%20DAIKIN%202012/4_Heating-Daikin_Altherma/2_LT_Monobloc/0_Archive/1_E%28D,B%29LQ_Monobloc/Documents/Databooks/EEDEN09-720_tcm135-130940.pdf [accessed 21 November 2013]

Daikin Altherma Bochure (PCAWUSE11-06B) [online] http://www.daikinac.com/content/assets/DOC/Product%20Brochures/PCAWUSE11-06B%20-%20Altherma%20Brochure%20-%20Daikin.pdf [accessed 21 November 2013]

Daikin *EKSOLHW plate heat exchanger* [online] http://www.daikin.co.uk/mdm/pdf/EKSOLHWAV1-EKHWS_dim_3TW57844-1_EN.pdf [accessed 21 November 2013]

Daikin. (2009) *Altherma Technical Data EEDEN09-720 - 03/2009.* [online] http://termopompa.ovo.bg/TD_Katalozi/TD_Termopompi_DAIKIN_Altherma.pdf [accessed 5 September 2013]

Danfoss. *Automation of commercial heating controls*. [online] http://www.danfoss.com/BusinessAreas/RefrigerationAndAirConditioning/EducationAndTraining/Automation+of+Commercial+Refrigeration+Plant.htm [accessed 4 September 2013]

DECC (2012) *The Future of Heating: A strategic framework for low carbon heat in the UK* [online] https://www.gov.uk/government/uploads/system/uploads/attachment_data/file/48574/4805-future-heating-strategic-framework.pdf [accessed 4 September 2013]

DECC, (2009) *The UK Supply Curve for Renewable Heat. Study for the Department of Energy and Climate Change* [online] http://webarchive.nationalarchives.gov.uk/20110123082441/decc.gov.uk/en/content/cms/what_we_do/uk_supply/energy_mix/renewable/policy/incentive/incentive.aspx [accessed 18 September 2011]

DECC, (2011) *Planning our electric future: a white paper for secure, affordable and low carbon electricity* [online] http://www.decc.gov.uk/assets/decc/11/policy-legislation/emr/2210-emr-white-paper-full-version.pdf [accessed 18 September 2011]

DECC, (2011) *Renewable Heat Initiative* [online] http://www.decc.gov.uk/assets/decc/What%20we%20do/UK%20energy%20supply/Energy%20mix/Renewable%20energy/policy/renewableheat/1387-renewable-heat-incentive.pdf [accessed 18 September 2011]

DECC, (2011) *The Carbon Plan* [online] http://www.decc.gov.uk/assets/decc/What%20we%20do/A%20low%20carbon%20UK/1358-the-carbon-plan.pdf [accessed 18 September 2011]

DECC, (2012) *Microgeneration Installation Standard: MIS 3005. Requirements for contractors undertaking the supply, design, installation, set to work commissioning and handover of microgeneration heat pump systems*. Issue 3.1a. [online] http://www.microgenerationcertification.org/images/MIS_3005_Issue_3.1a_Heat_Pump_Systems_2012_02_20.pdf [accessed 30 September 2012]

DECC, *Renewable Heat Premium Payment scheme* [online] https://www.gov.uk/renewable-heat-premium-payment-scheme [accessed 8 September 2013]

DECC. (2012) *How the Green Deal will reflect the in-situ performance of energy efficiency measures*. [online] https://www.gov.uk/government/uploads/system/uploads/attachment_data/file/48407/5505-how-the-green-deal-will-reflect-the-insitu-perfor.pdf [accessed 6 September 2013]

DECC. (2012) *MIS 3005 Requirements for contractors undertaking the supply, design, installation, set to work commissioning and handover of microgeneration heat pump systems.* [online] http://www.microgenerationcertification.org/images/MIS%203005%20Issue%203.2%20Heat%20Pump%20Systems%202013.07.22.pdf [accessed 5 September 2013]

DECC/BRE (2011) SAP 2009, *The Government's standard assessment procedure for energy rating of dwellings*. [online] http://www.bre.co.uk/filelibrary/SAP/2009/SAP-2009_9-90.pdf [accessed 5 September 2013]

DEFRA, (2011) *Guidelines to Defra / DECC's GHG Conversion Factors for Company Reporting* [online] http://archive.defra.gov.uk/environment/business/reporting/pdf/110707-guidelines-ghg-conversion-factors.pdf [accessed 18 September 2011]

Delta Energy and Environment (2011) *Heat Pumps in the UK: How Hot Can They Get?* [online] http://www.sepemo.eu/fileadmin/red/Publications/Delta_Heat_Pump_Trials_Whitepaper_January_2011.pdf [accessed 18 September 2011]

Department of Homeland Security Science and Technology Directorate, (2009) *Department of homeland Security science and technology readiness level calculator (ver 1.1) Final Report and user's manual* [online] http://www.homelandsecurity.org/hsireports/DHS_ST_RL_Calculator_report20091020.pdf [accessed 25 September 2011]

Dimplex SI 11TE, Ref: 452232.66.12. *Installation and Operating Instructions*. [online] http://www.dimplex.de/downloads/uploads/si5-21te_fd8611_gb-2007-05-15.pdf [accessed 5 September 2013]

Dimplex. (2010) *Smartrad planning manual*. [online] http://www.dimplex.co.uk/assets/Downloads_Documents/SmartRad_planning_manual.pdf [accessed 5 September 2013]

Dowson, M. (2012) *Passivhaus refurbishment* in *Carbon Bites*, CIBSE Energy Performance Group. [online] http://www.cibse.org/content/microsites/epg/cb18.pdf [accessed August 2013]

DuPont refrigerant data sheets (online) http://www2.dupont.com/Refrigerants/en_GB/products/msds.html [accessed 15 December 2013)

Du Toit, S. H. C., Steyn, A. G. W., Stumpf, R H. (1986) *Graphical Data Analysis*. Springer-Verlag

Dunbabbin, P., Wickins, C. (2012) *Detailed Analysis from the first phase of the Energy Saving Trust's heat pump trial: April 2009 to April 2010*. DECC. [online] http://www.decc.gov.uk/assets/decc/11/meeting-energy-demand/microgeneration/5045-heat-pump-field-trials.pdf [accessed 5 September 2013]

Dunbabbin, P., Charlick, H., Green, R. (2013) *Detailed analysis from the second phase of the Energy Saving Trust's heat pump field trial*. [online] https://www.gov.uk/government/uploads/system/uploads/attachment_data/file/225825/analysis_data_second_phase_est_heat_pump_field_trials.pdf [accessed 5 September 2013]

EHPA EUCERT-HP. [online] http://www.ehpa.org/european-certified-hp-installer/ [accessed 5 September 2013]

EHPA, (2009)[a]. *EHPA Testing Regulation. Testing of Air/Water Heat Pumps. Terms, Test Conditions and Test Method based on EN14511-1 through 4. Supplemental requirements for granting the international quality label for heat pumps.* Version 1.3 Release 24.03.2009. [online] http://www.ntb.ch/fileadmin/Institute/IES/pdf/EHPA-DACH%20TestReg%20AW-HP%20V1.3.pdf [accessed 6 September 2013]

EHPA, (2009)[b]. *EHPA Testing Regulation. Testing of Water/Water and Brine/Water Heat Pumps. Terms, Test Conditions and Test Method based on EN 14511-1 through 4. Supplemental requirements for obtaining the international quality label for heat pumps.* Version 1.3 Release 24.03.2009. [online] http://www.ntb.ch/fileadmin/Institute/IES/pdf/EHPA-DACH%20TestReg%20BW-WW-HP%20V1.3.pdf [accessed 6 September 2013]

EHPA, (2009)[c]. *EHPA Testing Regulation. Testing of Heat Pumps for Domestic Hot Water Production. Terms, Test Conditions and Test Method based on EN 255 Part 3. Supplemental requirements for granting the international quality label for heat pumps*. Version 1.0 Release 24.03.2009. [online] http://www.ntb.ch/fileadmin/Institute/IES/pdf/EHPA-DACH%20TestReg%20DHW-HP%20V1.0.pdf [accessed 6 September 2013]

Eljidi, A. (2011) *BSRIA UK heating market*. [online] http://www.r744.com/articles/141520110609.php also http://www.ehpa.org/fileadmin/red/EHPA_Activities/4th_Heat_Pump_Forum_2011/04_HPForum_UK_Heating_Market_2011.pdf [accessed 5 September 2013]

Energy Saving Trust and Defra. (2008) *Measurement of domestic hot water consumption in dwellings.* [online] https://www.gov.uk/government/uploads/system/uploads/attachment_data/file/48188/3147-measure-domestic-hot-water-consump.pdf [accessed 5 September 2013]

Energy Saving Trust (2008) *Consultation Responses for the Draft Detailed Technical Specification for Field Monitoring of Heat pump installations for residential dwellings, January/February 2008.*

Energy Saving Trust. (2009) *Final Report: Insitu monitoring of efficiencies of condensing boilers and use of secondary heating*. [online] https://www.gov.uk/government/uploads/system/uploads/attachment_data/file/180950/In-situ_monitoring_of_condensing_boilers_final_report.pdf [accessed 6 September 2013)

Energy Saving Trust. (2010) *Getting warmer, a field trial of heat pumps*. http://www.energysavingtrust.org.uk/Media/node_1422/Getting-warmer-a-field-trial-of-heat-pumps-PDF [accessed 18 September 2011]

Energy Saving Trust. (2010) *Heat Pump Field Trial Technical report*. Confidential

Energy Saving Trust (2013) *The heat is on: heat pump field trials: phase 2*. [online] http://www.energysavingtrust.org.uk/Organisations/Working-with-Energy-Saving-Trust/The-Foundation/Our-pioneering-research/The-heat-is-on-heat-pump-field-trials [accessed 5 September 2013]

Energy Saving Trust *Renewable Heat Incentive*. [online] http://www.energysavingtrust.org.uk/Generating-energy/Getting-money-back/Renewable-Heat-Incentive-RHI [accessed 5 September 2013]

Erb, M., Hubacher, P., Ehrbar, M. (2004) *Feldanalyse von Wärmepumpenanlagen FAWA 1996-2003.* EnergieSchweiz. [online] http://www.bfe.admin.ch/dokumentation/energieforschung/index.html?lang=en&publication=8070 [accessed 30 September 2012]

Eschmann, M. (2004) *Auswertung und analysen von Klein-Wärmepumpen. Interstaatliche Hochschule für Technik NTB.* [online] http://www.bfe.admin.ch/suchen/index.html?keywords=eschmann&go_search=suchen&lang=

de&site_mode=intern&nsb_mode=yes&search_mode=AND#volltextsuche [accessed 30 September 2012]

EUCERT *Heat pump installer manual, UK Edition, EE04.* Available from ConstructionSkills, UK.

European Commission Climate Action website [online] (http://ec.europa.eu/clima/policies/package/index_en.htm [accessed 18 September 2011]

European Commission, (1992) *CIL DIRECTIVE 92/42/EEC on efficiency requirements for new hot-water boilers fired with liquid or gaseous fuels.* [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=CONSLEG:1992L0042:20080321:EN:PDF [accessed 18 September 2011]

European Commission, (2002) *Mandate to CEN and CENELEC for the elaboration and adoption of measurement standards for household appliances: Water-heaters, hot water storage appliances and water heating systems.* [online] www.annex28.net/pdf/m324EN.doc [accessed 18 September 2011]

European Commission, (2005) DIRECTIVE 2005/32/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 6 July 2005 *establishing a framework for the setting of ecodesign requirements for energy-using products* and amending Council Directive 92/42/EEC and Directives 96/57/EC and 2000/55/EC of the European Parliament and of the Council [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2005:191:0029:0029:EN:PDF [accessed 18 September 2011]

European Commission, (2009) DIRECTIVE 2009/125/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 21 October 2009 *establishing a framework for the setting of ecodesign requirements for energy-related products*. [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2009:285:0010:0035:en:PDF [accessed 18 September 2011]

European Commission, (2009) *DIRECTIVE 2009/28/EC OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 23 April 2009 on the promotion of the use of energy from renewable sources and amending and subsequently repealing Directives 2001/77/EC and 2003/30/EC* [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2009:140:0016:0062:en:PDF [accessed 18 September 2011]

European Commission, (2010) DIRECTIVE 2010/31/EU OF THE EUROPEAN PARLIAMENT AND OF THE COUNCIL of 19 May 2010 on the energy performance of buildings [EPBD]. [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2010:153:0013:0035:EN:PDF [accessed 6 September 2013)

European Commission, (2010) *COMMUNICATION FROM THE COMMISSION TO THE EUROPEAN PARLIAMENT, THE COUNCIL, THE EUROPEAN ECONOMIC AND SOCIAL COMMITTEE AND THE COMMITTEE OF THE REGIONS. Analysis of options to move beyond 20% greenhouse gas emission reductions and assessing the risk of carbon leakage* [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=COM:2010:0265:FIN:EN:PDF [accessed 18 September 2011]

European Commission. (2012) *Guidance Document: The Monitoring and Reporting Regulation – Guidance on Uncertainty Assessment. MRR Guidance document No. 4*, Final Version of 5 October 2012. [online] http://ec.europa.eu/clima/policies/ets/monitoring/docs/gd4_guidance_uncertainty_en.pdf [accessed 20 May 2014]

European Commission. (2013) *Commission decision of 1 March 2013 establishing the guidelines for Member States on calculating renewable energy from heat pumps from different heat pump technologies pursuant to Article 5 of Directive 2009/28/EC of the European Parliament and of the Council*. [online] http://eur-lex.europa.eu/LexUriServ/LexUriServ.do?uri=OJ:L:2013:062:0027:0035:EN:PDF [accessed 6 September 2013]

European Heat Pump News. (2002) *Education and Training Committee in the EHPA*. Issue 4/3, December 2002. [online] http://www.pac.ch/dateien/EHPN_Issue_4=3.pdf [accessed 5 September 2013]

Eurostat (2013) η *(eta) time series* [online] http://epp.eurostat.ec.europa.eu/portal/page/portal/energy/other_documents [accessed 14 November 2013]

Faulkner, H. (2008) *Appendix 7: Lot 11 - 'Circulators in buildings'*. Report to the European Commission. ED Number 02287, Issue Number 5. [online]

http://www.ebpg.bam.de/de/ebpg_medien/011_studyf_08-04_circulators_updated.pdf [accessed 4 September 2013]

Flyvbjerg, B. (2006) *Five Misunderstandings About Case-Study Research*, Qualitative Inquiry 12 (2) 219-245. (p221) [online] http://flyvbjerg.plan.aau.dk/Publications2006/0604FIVEMISPUBL2006.pdf [accessed 4 September 2013]

Geels, F. W. (2005) *Technological Transitions and System Innovations: A Co-evolutionary and Socio-Technical Analysis*, Cheltenham: Edward Elgar)

GeoEnergy. (2012) (MIS) *GSHP Hydraulics design guide*: *Procedure and charts for designing the hydraulics and associated pumping power of closed loop GSHP systems under MCS.* [online] http://www.microgenerationcertification.org/admin/documents/GSHP%20Hydraulics%20Design%20Guide%20v1.0.pdf [accessed 5 September 2013]

Geotrainet. [online] http://www.geotrainet.eu/moodle/ [accessed 6 September 2013]

Gledhill. *Stainless light buffer store, stainless steel store for use with heat pumps and boilers* (2013) . [online] http://www.gledhill.net/documents/StainlessLite_Buffer_Store_Manual.pdf [accessed 16 August 2013]

Gleeson, C. P., Clarke, L. (2013) *The neglected role of labour in low energy construction: 'thermal literacy' and the difference between design intention and performance.* Conference paper for Work in a Warming World: Labour, Climate Change and Social Struggle. Toronto, Canada. November 29 to December 1, 2013

Gleeson, C. P., Lowe, R. (2013) *Meta-Analysis of European Heat Pump Field Trial Efficiencies.* Energy & Buildings, 66, November 2013, pp637–647

Gleeson, C., Yang, J., Lloyd-Jones, T. (2011) *European Retrofit Network: Retrofitting Evaluation Methodology Report* [online] http://www.westminster.ac.uk/__data/assets/pdf_file/0003/108786/UW-ERN-Report-271011.pdf [accessed 6 September 2013]

Green Building Forum (online) http://www.greenbuildingforum.co.uk/newforum/comments.php?DiscussionID=895 [accessed 5 September 2013]

Griggs, P., McCall, M. (1988) *Domestic heat pumps: performance and economics.* BRE Report 126.

Grundfos, Ref: V7089107. *Grundfos data booklet, Magma, UPE. Series 2000 circulator pumps*. [online] http://net.grundfos.com/doc/webnet/magna/downloads/Magna_databooklet.pdf [accessed 5 September 2013]

Health and Safety Executive (online) http://www.hse.gov.uk/legionnaires/things-to-consider.htm [accessed 5 September 2013]

Health and Safety Executive (1998) Safety in the installation and use of gas systems and appliances. Gas Safety (Installation and Use) Regulations 1998. Approved Code of Practice and guidance. [online] http://www.hse.gov.uk/pubns/priced/l56.pdf [accessed 5 September 2013]

Heat King. Technical manual, 70408043-08. [online] http://www.heatking.co.uk/pdfs/Heatking_Technical_Manual_70408043-08.pdf [accessed 5 September 2013]

Highgate Society (undated) *Making heating systems efficient and cost-effective* [online] www.highgatesociety.com/modules/download_gallery/dlc.php?file=11 [accessed 14 November 2014]

Hochschule Bremen: Strauss, R-P., Seebörger, T. (2008) *Prüfbericht Prüfung eines Kompaktgerätes zur Zertifizierung als Passivhaus geeignete Komponente*. 'Consideration of a test report for certification as a compact unit as a sui4 Passivhaus component'. Available from Drexel & Weiss.

Hollingshead (2012) *Solar power incentives lose their shine*. The Daily Telegraph (04 02 2012) [online] http://www.telegraph.co.uk/comment/9059878/Solar-power-incentives-lose-their-shine.html [accessed 18 September 2011]

Honeywell Sundial. [online] http://www.honeywelluk.com/products/Systems/Wired-Sundial/ [accessed 5 September 2013]

Hubacher, P. (2004) *Field analysis of heat pump installations – the FAWA Project*. IEA Heat Pump Centre Newsletter. Volume 22, No 2/2004. [online] http://www-v2.sp.se/hpc/publ/HPCOrder/ViewDocument.aspx?RapportId=137 [accessed 30 September 2012]

Huber, H., Glasner, G. (2007) *IEA Heat Pump Centre Newsletter*. Volume 25 - No. 2/2007. [online] http://www.heatpumpcentre.org/en/newsletter/previous/Documents/HPC-news_2_2007.htm [accessed 30 September 2012]

IEA *IEA heat pump Annex 37 Demonstration of field measurements of heat pump systems in buildings - Good examples with modern technology* [online] http://www.heatpumpcentre.org/en/projects/ongoingprojects/annex37/Sidor/default.aspx [accessed 18 September 2011]

Institute for Value Management (IVM) *Value management techniques* [online] http://ivm.org.uk/techniques [accessed 14 November 2013]

Jahresarbeitszahlen der Wärmepumpen (online image) http://www.jahresarbeitszahlen.info/index.php/jahresarbeitszahl/systemgrenzen [accessed 30 September 2012]

JB Consulting International. *Technology Readiness Levels* [online] http://jbconsultinginternational.com/TechnologyReadinessLevel.aspx [accessed 25 September 2011]

Johnson, E, P. (2010) *Air-source heat pump carbon footprints: HFC impacts and comparison to other heat sources*. Energy Policy, 39 (2011) pp1369–1381

Jones, D. (2013) Personal correspondence with Danny Jones (16/07/2013) training consultant to the MCS heat pump working group.

Kensa Engineering. *Fact Sheet - Sizing Kensa Heat Pumps - 01* (online) http://www.kensaengineering.com/Library/Fact-sheets/New%20Fact%20Sheets/Fact%20Sheet%20-%20Sizing%20Kensa%20Compact%20Heat%20Pumps-01.pdf [accessed 5 September 2013]

Kensaheatpumps [sic] (online)
http://www.kensaengineering.com/installer/faq_detail.asp?id=65 [accessed 5 September 2013]

Kershaw, K., Lelyveld, T., Burton, S., Orr, G., Charlick, H., Dennish, T., Crowther, M. (2010) *In-situ monitoring of efficiencies of condensing boilers –TPI control project extension'* [online] http://www.decc.gov.uk/assets/decc/what%20we%20do/supporting%20consumers/saving_energy/analysis/1149-condensing-boilers.pdf [accessed 18 September 2011]

Lenz, J, R. (2002) Polytropic exponents for common refrigerants. Purdue-e-Pubs [online] http://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=2527&context=icec [accessed 17 December 2013]

Levermore, G. (1997) personal correspondence with Geoff Levermore,, University of Manchester, regarding controls and modelling central heating systems

Levine, M., D. Ürge-Vorsatz, K. Blok, L. Geng, D. Harvey, S. Lang, G. Levermore, A. Mongameli Mehlwana, S. Mirasgedis, A. Novikova, J. Rilling, H. Yoshino, (2007) *Residential and commercial buildings. In Climate Change 2007: Mitigation. Contribution of Working Group III to the Fourth Assessment Report of the Intergovernmental Panel on Climate Change* [B. Metz, O.R. Davidson, P.R. Bosch, R. Dave, L.A. Meyer (eds)], Cambridge University Press, Cambridge, United Kingdom and New York, NY, USA. [online] http://www.ipcc.ch/pdf/assessment-report/ar4/wg3/ar4-wg3-chapter6.pdf [accessed 18 September 2011]

Lowe, R. J. (2007) *Addressing the challenges of climate change for the built environment* (editorial), Building Research & Information, 35 (4) 343-350.

Lowe, R. (2007) *Technical options and strategies for decarbonizing UK housing*. Building Research and Information, 35 (4), pp412 – 425.

Lowe, R., Bell, M. (1998) *Towards Sustainable Housing: Building Regulation for the 21st Century*, Leeds Metropolitan University Centre for the Built Environment for Joseph Rowntree Foundation. [online]
http://www.leedsmet.ac.uk/as/cebe/projects/towards_sustainable_housing.pdf [accessed 4 September 2013]

Maier, A. *Refrigeration Basics: Optimizing System Performance Using a TXV*. [online] http://www.mcmech.ca/technician-resource/service-clinic-tx-valve-basics/html [accessed 4 September 2013]

Market update (2008). [online] http://www.centralheating.co.uk/system/uploads/attachments/0000/1757/August_08.pdf [accessed 18 September 2011]

Market Update (2009). [online] http://www.centralheating.co.uk/system/uploads/attachments/0000/0154/Feb_09_Market_Update.pdf [accessed 8 September 2013]

Markus, T. A., Morris, E. N. (1980) Buildings, climate and energy. Pitman, London

Martin, C., Watson, M. (2008) In Situ Measurement of the Performance of Heat Pumps in Dwellings: A technical specification for field monitoring.

Microgeneration scheme. [online] http://www.microgenerationcertification.org/ [accessed 5 September 2013]

MCS/DECC. (2008) *Micro Installation Standard: MCS 001 issue 2.2. Installer certification scheme requirements* [online] http://www.microgenerationcertification.org/images/MCS%20001%20-%20Issue%202.2%20Installer%20Certification%20Scheme%20Requirements%201st%20April%202013.pdf [accessed 5 September 2013]

MCS, (2009) *MCS Product Certification Scheme for heat pumps, MCS 007* [online] http://www.microgenerationcertification.org/admin/documents/MCS%20007%20-%20Issue%202%200%20Product%20Certification%20Scheme%20Requirements%20-%20Heat%20Pumps%2015%20Dec%2009.pdf [accessed 4 September 2013]

MCS. (2011) *Working Group Terms of Reference, 10/01/2011*. [online] http://www.microgenerationcertification.org/images/MCS_WG_ToR_v1.3.pdf [accessed 5 September 2013]

MCS (2012) *Heat Pump Working Group minutes, 13/12/2012*. [online] http://www.microgenerationcertification.org/images/FINAL%20Heat%20Pumps%20Working%20Group%20Minutes%20-%2013%20December%202012%20v0.3%20For%20general%20release.pdf [accessed 5 September 2013]

MCS *Heat pump software* (online) http://www.microgenerationcertification.org/mcs-standards/installer-standards/mcs-heat-pump-software [accessed 5 September 2013]

MCS (2013) *Installer certification scheme requirements*. [online] http://www.microgenerationcertification.org/images/MCS%20001%20-%20Issue%202.3%20Installer%20Certification%20Scheme%20Requirements.pdf [accessed 20 December 2013]

MCS *Statistics*. (2013) [online] http://www.microgenerationcertification.org/about-us/statistics [accessed 8 September 2013]

MCS. (2013) *Microgeneration Installation Standard: MIS 3005 issue 3.2* [online] http://www.microgenerationcertification.org/images/MIS%203005%20Issue%203.2%20Heat%20Pump%20Systems%202013.07.22.pdf [accessed 5 September 2013]

Miara, M, Gunther, D., Kramer, T., Oltersdorf, T. Wapler, J. (2011)[a] *Wärmepumpen Effizienz, Messtechnische Untersuchung von Wärmepumpenanlagen zur Analyse und Bewertung der Effizienz im realen Betrieb.* [online] http://wp-effizienz.ise.fraunhofer.de/download/wp_effizienz_endbericht_langfassung.pdf [accessed 30 September 2012]

Miara, M, Gunther, D., Kramer, T., Oltersdorf, T. Wapler, J. (2011)[b] *Heat Pump Efficiency, Analysis and Evaluation of Heat Pump Efficiency in Real-life Conditions.* Fraunhofer. http://wp-effizienz.ise.fraunhofer.de/download/final_report_wp_effizienz_en.pdf [accessed 30 September 2012]

Mitsubishi Electric Eco-cute (online) http://www.mitsubishielectric.com/company/environment/report/product_info/appliances/heatpump/index.html [accessed 24 November 2013]

Morgan, R. (2010) *Green boiler scheme could be scrapped over reliability fears*. The Daily Telegraph (31 08 2010) [online] http://www.telegraph.co.uk/finance/newsbysector/utilities/7972003/Green-boiler-scheme-could-be-scrapped-over-reliability-fears.html [accessed 18 September 2011]

Moss, K. (2003) *Heating and hot water design in buildings*. 2nd Edition. Spon Press

Myatt, G. J. (2007) *Making Sense of Data A practical guide to exploratory data analysis and data mining* Wiley-Interscience

Myson. (2012) *Panel radiators, technical guide*. [online] http://www.myson.co.uk/static_files/my/media/downloads/Panel_Tech_Guide_-_August_2012%281%29.pdf [accessed 5 September 2013]

Nibe (undated) Nibe *Fighter 360P. Installation and maintenance, MOS GB 0418-1 411482*. [online] http://www.nibeonline.com/pdf/411482-1.pdf [accessed 5 September 2013]

Nordman, R., Lindahl, M., Anderson, K., Miara, M., Ceovet, M., Kleefkens, O., Planje, W., Zottl, A., Polyzou, O., Benou, A., Karytsas, C. (2012) *D2.1 Survey of completed and ongoing field measurements of heat pumps with hydronic heating systems.* [online] http://www.sepemo.eu/uploads/media/D2_1_Survey_of_completed_and_ongoing_field_measurements_v2.pdf [accessed 6 September 2013)

Nordman, R., Andersson, K., Axell, M., Lindahl, M. (2010) *Calculation methods for SPF for heat pump systems for comparison, system choice and dimensioning. SP Report 2010:49.* SP Technical Research Institute of Sweden. [online] http://www.sp.se/en/index/services/heatpump/sidor/default.aspx [accessed 30 September 2012]

Nørgard, J,S., Holck, J., Mehlsen, K. (1983) *Langsigtede tekniske muligheder for el-besparelser*. [Long-term technical potential for electricity savings] Fysick Laboratorium III, Danmarks Tekniske Højskole, 2800 Lyngby.

Nowak, T. (2011) *Heat pumps – a renewable energy technology?* REHEV Journal. [online] www.rehva.eu/download/_/441/rj4_10-12.pdf [accessed 30 September 2012]

Nowak, T. (2012) *European Heat Pump Association predicts market recovery in 2012*. [online] http://www.rehva.eu/fileadmin/hvac-dictio/05-2012/p53-54_nowak.pdf [accessed 8 September 2013]

Office of the Deputy Prime Minister, (ODPM), (2005) *Gas and oil central-heating boilers, Advice to householders* [online] http://www.planningportal.gov.uk/uploads/br/BR_PDF_PTL_GASHEATADVICE.pdf [accessed 14 November 2013]

OFQUAL[a] (2012) *BPEC Level 2 NVQ Diploma in Plumbing and Heating* (QCF) (online) http://register.ofqual.gov.uk/Qualification/Details/600_5270_3 [accessed 5 September 2013]

OFQUAL[b] (2012) *BPEC Level 3 NVQ Diploma in Plumbing and Heating* (QCF)[online] http://register.ofqual.gov.uk/Qualification/Details/600_6863_2 [accessed 5 September 2013]

OFQUAL (2010). *Functional skills* [formally "key skills"]. [online] http://www2.ofqual.gov.uk/qualifications-assessments/89-articles/15-functional-skills [accessed 6 September 2013]

Open University, 2010. *Heat Pump user experiences, behaviour, perceptions and satisfaction.* EST Field trial documents. Confidential

Oreszczyn, T., Lowe, R. (2009) *Challenges for energy and buildings research: objectives, methods and funding mechanisms*. Building Research & Information, 38:1, pp107-122

Orr, G., Lelyveld, T., Burton, S., Summerfield, I. (2009) *In-situ monitoring of efficiencies of condensing boilers and use of secondary heating.* Energy Savings Trust Report. [online] http://www.energysavingtrust.org.uk/Publications2/Housing-professionals/Heating-systems/In-situ-monitoring-of-efficiencies-of-condensing-boilers-and-use-of-secondary-heating-trial-final-report [accessed 30 September 2012]

Oxford dictionaries. [online] http://www.oxforddictionaries.com/ [accessed 7 May 2014]

Passivhaus. Drexel und Weiss compact unit certificate. [online] http://www.passiv.de/komponentendatenbank/files/pdf/zertifikate/zd_drexel_weiss_aerosmartm_de.pdf [accessed 6 September 2013]

Passivhaus Institute, (2007) *Prüfverfahren zur energetischen und schalltechnischen Beurteilung von Wärmepumpen-Kompaktgeräten für die Zertifizierungals "Passivhaus geeignete Komponente"* 'Test methods for energetic and sound technical evaluation of heat pump compact units for certification as a "Passive suitable component"'. [online] http://www.passiv.de/old/03_zer/Komp/Komp/Pruefreglement_K.pdf [accessed 6 September 2013]

Passivhaus "PH*Luft"* software. [online] http://www.passiv.de/de/05_service/02_tools/03_phluft/03_phluft.htm [accessed 6 September 2013]

Pederson, S., Jacobsen, E. (2011) *Approval of systems entitled to subsidies. Measurements, data collection and dissemination.* Danish Technological Institute. Available from the authors.

Philliber, S. G., Schwab, M. R., & Samsloss, G. (1980) *Social Research: Guides to a decision making process*. Itasca, IL: Peacock

Poverty site. [online] http://www.poverty.org.uk/s77/index.shtml [accessed 18 September 2011]

Pressure-Volume diagram. [online] http://upload.wikimedia.org/wikipedia/commons/thumb/0/06/Carnot_cycle_p-V_diagram.svg/400px-Carnot_cycle_p-V_diagram.svg.png [accessed 4 September 2013]

RenEnergy (online) http://www.renenergy.co.uk/home-gshp.aspx [accessed 5 September 2013]

Renewable Heat Incentive. [online] https://www.gov.uk/government/policies/increasing-the-use-of-low-carbon-technologies/supporting-pages/renewable-heat-incentive-rhi [accessed 6 September 2013]

Renewable Heat Premium Payment scheme. [online] https://www.gov.uk/renewable-heat-premium-payment-scheme [accessed 6 September 2013]

Rogers, G. R., Mayhew. Y. (1992) *Engineering thermodynamics, work and heat transfer*. 4th Edition. Longman Scientific & Technical

Rosenfeld, Arthur H. (1999) *The Art of Energy Efficiency: Protecting the Environment with Better Technology, Annual Review of Energy and the Environment*, 24 33-82 p69.

Russ, C., Miara, M., Platt, M., Günther, D., Kramer, T., Dittmer, H., Lechner, T., Kurz, C. (2010) *Feldmessung Wärmepumpen im Gebäudebestand* (Heat pump field trial in existing buildings). Fraunhofer. [online] http://www.wp-im-gebaeudebestand.de/download/WP_im_Gebaeudebestand_Kurzfassung.pdf [accessed 30 September 2012]

Sanyo CO2 ECO Heating System (online) http://www.acr-news.com/products/product.asp?id=1029 [accessed 24 November 2013]

Scottish Sector Profile. (2011) *Alliance of Sector Skills Councils*. [online] http://www.researchonline.org.uk/sds/search/download.do?ref=LMD947 [accessed 5 September 2013]

SEDBUK website. [online] http://www.sedbuk.com/cgi-local/dynamicv.cgi?page=boiler5 [accessed 18 September 2011]

SEPEMO-Build. [online] http://www.sepemo.eu/ [accessed 30 September 2012]

Singh, A. *Optimum control with E2™*. [online] http://www.emersonclimate.com/en-us/WhitePapers/Optimum-Refrigeration-Control-with-E2.pdf [accessed 4 September 2013]

Staffell, I. (2009) *A Review of Domestic Heat Pump Coefficient of Performance*. [online] http://imperial.academia.edu/IainStaffell/Papers/1126238/A_review_of_domestic_heat_pump_coefficient_of_performance [accessed 30 September 2012]

Stafford, A. (2011) *Long term monitoring and performance of ground-source heat pumps. Building Research and Information*, 39:6, pp566-573.

Stafford, A., Lilley, D. (2012) *An investigation into a single ground-source heat pump in the context of 10 similar systems*. Energy and Buildings, 49 pp536-541.

Stake, R. E. (1995) *The Art of Case Study Research*. Sage Publications

STATA Data analysis and statistical software [online] www.stata.com [accessed 25 September 2011]

Statistics for Wales (2008) *Living in Wales 2004 – Heating and Energy Measures.* [online] http://wales.gov.uk/docs/statistics/2005/051202sdr1352005ren.pdf [accessed 18 September 2011]

Stenlund, M., Axell, M. (2007) *Residential ground source heat pump systems – results from a field study in Sweden*. Available online: www.annex32.net/pdf/articles/GSHP%20article_SP_2010.pdf [accessed 30 September 2012]

Tukey, J. W. (1977) *Exploratory Data Analysis* Addison-Wesley Publishing Company

UK Committee on Climate Change. (2010) *The Fourth Carbon Budget - reducing emissions through the 2020s* [online] http://www.theccc.org.uk/reports/fourth-carbon-budget [accessed 25 September 2011]

UK GSHPA (online) http://www.gshp.org.uk/contact_GSHPA.html [accessed 5 September 2013]

Vaughan, A. (2010) *UK heat pumps fail as green devices, finds study*. The Guardian (01 08 2010) [online] http://www.guardian.co.uk/environment/2010/sep/08/heat-pumps-green-heating [accessed 18 September 2011]

VDI, 2009. *VDI 4650 Blatt 1*: *Berechnungen von Wärmepumpen - Kurzverfahren zur Berechnung der Jahresarbeitszahl von Wärmepumpenanlagen - Elektro-Wärmepumpen zur Raumheizung und Warmwasserbereitung.* 'Calculation of heat pumps - Simplified method for the calculation of the seasonal performance factor of heat pumps - Electric heat pumps for space heating and domestic hot water'.

Velleman, P, F., Hoaglin, D, C. (2004) *Applications basics and computing of exploratory data analysis*. [online] http://dspace.library.cornell.edu/handle/1813/78 [accessed 5 September 2013]

Wärmepumpen Testzentrum (2009) *WPZ Bulletin, Ausgabe 01-2009, Informationsblatt des Wärmepumpen-Testzentrums Buchs. Grosser Unterschied zwischen EN 14511 und EN 255* [online] http://institute.ntb.ch/fileadmin/Institute/IES/pdf/WPZ%20Bulletin%2001-2009%20DE.pdf [accessed 23 November 2013]

Wemhoner, C., Afjai, T. (2003) *Seasonal performance calculation for residential heat pumps with combined space heating and water production (FHBB method)*. [online] http://www.bfe.admin.ch/php/includes/container/enet/flex_enet_anzeige.php?lang=de&publication=7926&height=400&width=600 [accessed 30 September 2012]

Wemhhoner. C, Afjai. Th. (2006) *Final Report IEA HPP Annex 28 Test procedure and seasonal performance calculation for residential heat pumps with combined space and domestic hot water heating. Part 1: Proposals for calculation method and test procedure*. Published by: IEA Heat Pump Centre, Box 857, SE-501 15 Borås, Sweden.

Wemhoner,C., Afjei, T., Dott, R. (2008) *IEA HPP Annex 28 - standardised testing and seasonal performance calculation for multifunctional heat pump systems,* p6.. Applied Thermal Engineering, 28, 2062-2069.

Whalley, P. B. (1992). *Basic engineering thermodynamics*. Oxford university press

Winch, C. (2012) *Skill - a concept manufactured in England*? From Brockmann et al, (2011) *Knowledge, Skills and Competence in the European Labour Market, What's in a vocational qualification*? Routledge, Oxon.

Wingfield, J., Bell, M., Miles-Shenton, D., South, T., Lowe, R. (2008) *Lessons from Stamford Brooke*. [online] http://www.leedsmet.ac.uk/as/cebe/projects/stamford/pdfs/stambrk-final-pre-pub.pdf [accessed 6 September 2013)

Worcester-Bosch. Greensource. User manual Air to water heat pump with domestic hot water distribution unit. 6 720 642 344 (2010/01). [online] http://www.worcester-bosch.co.uk/installer/literature/air-source-heat-pumps-literature/greensource-air-to-water-heat-pump-6kw-literature [accessed 5 September 2013]

Yin, R. K. (2009) *Case study research, Design and methods* 4th Edition. Sage Publications

Zottl, A., Nordman, R. (2011) *D4.2. / D 2.4. Concept for evaluation of SPF. Version 2.0. A defined methodology for calculation of the seasonal performance factor and a definition which devices of the system have to be included in this calculation. Heat pumps with hydronic heating systems*. [online] http://www.SEPEMO.eu/deliverables/wp4/ [accessed 21 November 2011]

Zottl, A., Nordman, R. (2012) *D4.2. / D 2.4. Concept for evaluation of SPF. Version 2.2. A defined methodology for calculation of the seasonal performance factor and a definition which devices of the system have to be included in this calculation. Heat pumps with hydronic heating systems*. [online] http://www.SEPEMO.eu/deliverables/wp4/ [accessed 30 September 2012]

Glossary

BPEC: "The BPEC group of companies specialise in providing operatives working in the building services engineering industry with the skills and expertise necessary to meet the high industry quality standards. Originally established in 1997, BPEC Certification Ltd. operates as both an Awarding Organisation and a Certification Body overseeing the competence assessment of individuals working in the industry." [online] http://www.bpec.org.uk/about/

Buffer vessel: A thermal store used to reduce heat pump cycling and usually feeding the just the space heating system.

Building Research Establishment: BRE. Formerly a UK Government research centre privatised in 1980 and specialising in building research, consultancy and testing.

Building Services Research and Information Association: BSRIA. A UK based provider of building services research, testing and consultancy. BSRIA represents the UK on the EUCERT-HP Education Committee.

Carnot efficiency: The theoretical maximum efficiency of a heat pump. For practical heat pumps, the Carnot efficiency is based on the thermodynamic temperature (Kelvin) of the thermal reservoirs rather than the temperatures of the evaporator and condenser. Thus the important role played by the emitter temperature as the only reservoir temperature over which the designer has total control.

Coefficient of Performance: COP. An instantaneous or cyclical measure of heat pump performance. Generally associated with either thermodynamic analysis or with manufacturers' technical data.

Combination boiler or 'combi': A boiler that combines both space heating and instantaneous domestic hot water within the same unit. Generally supplied as a 'system boiler', that is one with an expansion vessel, circulating pump and automatic controls for modulation of heat output. The combi negates the need for a separate hot water cylinder, a feed and expansion cistern and flow control valves. Most combis are supplied with a built in temperature controls option and diagnostic software.

Condensing boiler: A boiler designed to extract heat from the flue gases by condensing the water vapour resulting from combustion of hydro-carbons. The condensing function provides up to an additional 14% efficiency over non-condensing boilers. Condensing boiler efficiency is best expressed based on a gross calorific value as opposed to net calorific value.

Domestic hot water: DHW. Hot water for the use of occupants, generally for washing, as opposed to hot water for circulation through a space heating system of emitters.

Economy tariff: Off-peak electricity supplied at a cheaper rate. Commonly known as Economy 7 or Economy 10

EUCERT-HP. A European Heat Pump Association (EHPA) educational programme for designers and installers.

IEA: International Energy Agency

IPCC: Intergovernmental Panel on Climate Change. A United Nations body that provides updated reports on the impact of climate change and its mitigation.

Jahresarbeitzahlen: JAZ. Translated as Seasonal Performance Factor. German boundary definitions, see Table 5-1.

Micro-CHP: A combined heat and power unit designed for residential installation. CHP is based electricity generation with integrated heating supplied by the waste energy from the generator and/or the flue gases.

Microgeneration Scheme: MCS represents renewable energy stakeholders. Its role is to ensure registration of renewable technologies and to promote quality assurance and minimum training requirements.

Microgeneration Installation Standard: MIS. MCS installation standards for renewable technologies. The heat pump standard is MIS 3005.

Morphology: "The branch of biology that deals with the form of living organisms, and with relationships between their structures." (Oxford dictionaries, online)

NVQ: UK National Vocation Qualifications ranging from Level 1 (entry level) to Level 8 (doctorate).

Pathology: "The science of the causes and effects of diseases, especially the branch of medicine that deals with the laboratory examination of samples of body tissue for diagnostic or forensic purposes" (Oxford dictionaries, online)

Primary coil: The heat exchange coil inside the hot water cylinder.

Primary water: Water heated in the heat source and circulated through the primary flow and return to the primary coil in the hot water cylinder.

OFQUAL: UK Office of Qualifications and Examination Regulation. Governing body for NVQs. [online] http://ofqual.gov.uk/

Renewable Heat Incentive: RHI. A UK proposed scheme to pay residential occupants for the renewable heat provided by low and zero carbon technologies such as heat pumps, solar thermal and biomass boilers.

Renewable Heat Premium Payment Scheme: RHPP. A UK government project whereby building owners are paid an allowance for the installation of renewable heat technology. Under the RHPP only MCS registered companies may install heat pumps and the system must be monitored.

SAP: Standard Assessment Procedure. The UK methodology for assessing regulated building energy use for the purposes of Building Regulation, Approved Document ADL1a compliance.

SCOP: Seasonal coefficient of performance. An annual assessment of heat pump efficiency based on modelling COP values using the temperature bin method.

SCOPnet: Seasonal Coefficient of Performance without backup. May be used as an alternative to SPFH2 for RES Annex VII renewable heat assessment.

Secondary heating: In the context of the thesis, secondary heating relates to a living room or lounge fire that acts in support or as an alternative to central heating.

Secondary water: The DHW in the hot water cylinder.

Seasonal Performance Factor: SPF. An annual measurement of heat pump efficiency. The different definitions of SPF are based on which system components are included within the measured system boundary, see Table 5-1 for definitions.

SEDBUK: Seasonal Efficiency of Domestic Boilers UK. An online database of boiler efficiency.

SEFF: Seasonal efficiency of heat pumps based on the UK EST/DECC definition of system boundary, see Table 5-1 for definition.

SEPEMO: SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector. An EU 'Intelligent Energy" research project with the primary aim of establishing heat pump system monitoring standards. It has developed four boundary definitions for the purpose of measuring the SPF of installations.

Slinky: A ground loop heat exchange pipe supplied as a roll of plastic tubing designed to be laid either horizontally or vertically as loops rather than straight pipe. "'Slinky' is the term commonly used rather than the imposing scientific name - curtate cycloid. When used with geothermal heat pump systems, the Slinky is an overlapped plastic pipe circular coiled ground loop heat exchanger. It concentrates the heat transfer surface into a smaller volume, requiring shorter trenching". [on line]
http://c03.apogee.net/contentplayer/?coursetype=geo&utilityid=oge&id=7082

System boiler: A boiler supplied with a pressure vessel negating the need for an open vented feed and expansion cistern. The boiler can therefore be placed anywhere in the building irrespective of height above any cold water cisterns. Ideally suited to unvented hot water storage.

Taxonomy: "The branch of science concerned with classification, especially of organisms; systematics." (Oxford dictionaries, online)

STATA example ASHP data analysis 'Do-file"

```
// save imported file as a Stata file
save "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\426", replace
// open main database
use "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\ashp", clear
//check if 426 has already been added
qui count if ID == 426
if (r(N) == 0) {
 // add 426 data into main file
 append using "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\426"
 // save main database
 save "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\ashp", replace

sort ID time

// HW Flow Tot Time

table ID, contents(sum v19) // display table showing tot HW Flow

bys ID: egen HWFlowTot = sum(v19)

// SH Energy Tot Time

table ID, contents(sum v21) // display table showing tot SH Energy

by ID: egen SHEnergyTot = sum(v21)

// ASHP Energy Out

table ID, contents(sum v24) // display table showing tot ASHP Total Output

by ID: egen ASHPEnergyOut = sum(v24)

// Electrical input/Immersion

table ID, contents(sum v18) // display table showing tot Electrical input/Immersion

by ID: egen Immersion = sum(v18)

// System Boost

table ID, contents(sum v26) // display table showing tot WhSystem Boost
by ID: egen WhBoost = sum(v26)

// 2nd ASHP input

table ID, contents(sum v27) // display table showing 2nd ASHP input
by ID: egen HP2EnergyIn = sum(v27)

// Heat Pump Tot Power Input

table ID, contents(sum v16) // display table showing tot ASHP Tot Input

by ID: egen ASHPEnergyIn = sum(v16)

// Hot Water Energy

table ID, contents(sum v20) // display table showing tot DHW Tot Output

by ID: egen DHWEnergyOut = sum(v20)

// Primaries energy

table ID, contents(sum v28) // display table showing tot Primary F&R Energy

by ID: egen PrimaryEnergy = sum(v28)

// creating smaller file with info on each system only no time data
preserve
drop v* time
by ID: keep if _n == 1
save "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\ashp_byid", replace
restore
```

```
// generating efficiency scores
cap drop SPF
gen SPF = (DHWEnergyOut + SHEnergyTot) / (ASHPEnergyIn + Immersion)
// fix for any anomalies

// Removing no-relevant files.  Note these are not removed but marked as ZERO (0)
gen remove = 1

replace remove = 0 if ID == 477 |  ID == 483 |  ID == 489 |  ID == 490 | ID == 950 | ID == 927// Name v22 as ASHP out 2

bys ID: egen ASHPout2 = sum(v22)

// CLEARING ALL SPF ANOMALIES

replace SPF = (DHWEnergyOut + SHEnergyTot) / (ASHPEnergyIn + Immersion + WhBoost) if ID == 486 | ID == 487    // Change formula for the system 429, 447, 474, 475, 486, 487 to include v26 in efficienyc, this is an anomoly for this ID only

replace SPF = ASHPEnergyOut / (ASHPEnergyIn + WhBoost) if ID == 474 | ID == 475  // Change formula for systems 474 and 475 for these ASHPs only

replace SPF = ASHPEnergyOut / (ASHPEnergyIn + WhBoost) if ID == 447

replace SPF = (DHWEnergyOut + SHEnergyTot)/(ASHPEnergyIn + Immersion + WhBoost) if ID == 424 | ID == 486 | ID == 487 // include v26 in denominator

replace SPF = (DHWEnergyOut + SHEnergyTot)/(ASHPEnergyIn + HP2EnergyIn) if ID == 441  // include v27 in denominator

replace SPF = (DHWEnergyOut + SHEnergyTot)/ ASHPEnergyIn if ID == 440  //

replace SPF = (DHWEnergyOut + SHEnergyTot + PrimaryEnergy)/ ASHPEnergyIn if ID == 425  // Change formula for the system 425 to include v28 in efficiency, this is an anomoly for this ID only

replace SPF = SHEnergyTot / (ASHPEnergyIn + WhBoost) if ID == 479 | ID == 429

replace SPF = ASHPEnergyOut / (ASHPEnergyIn + WhBoost) if ID == 474 | ID == 475

replace SPF = ASHPEnergyOut / ASHPEnergyIn if ID == 442 | ID == 443  | ID == 444  | ID == 445  | ID == 446  | ID == 472

replace SPF = SHEnergyTot / (ASHPEnergyIn + Immersion) if ID == 478 // See notes on Field trial final report

// 426 spf

replace SPF = (DHWEnergyOut + SHEnergyTot) / (ASHPEnergyIn + Immersion) if ID == 426

//  Generate COP
gen COP = ASHPEnergyOut / ASHPEnergyIn

// fix for any anomalies
//418 has 2 heat pumps (EA and AS).  v24 (HP output) measures the output of the ASHP only

replace COP = 0 if ID == 418

replace COP = ASHPEnergyOut / (ASHPEnergyIn + HP2EnergyIn) if ID == 441  // Change formula for the system 441 to include v27 [28 = 27 in spreadsheet] in efficiency, this is an anomoly for this ID only

replace COP = ASHPEnergyOut / ASHPEnergyIn if ID == 426

// checking what we have generated
table ID, contents(min SPF min COP)

save "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\ashp_byid", replace
restore

//NEED TO REMOVE 477, 483, 489, 490
//477 is really GSHP with AS backup (doing virtually nothing)
//483, 489, 490 Air to Air with insufficient data for either COP or SPF

// SET COP AND SPF TO ZERO OR NEW FORMULA DUE TO SYSTEM SET UP

replace COP = 0 if ID == 418 | ID == 429 | ID == 474 | ID == 475 | ID == 483

replace SPF = (ASHPEnergyOut + WhBoost) / (ASHPEnergyIn + WhBoost) if ID == 447
```

```
save "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\ashp_bysyst_metered.dta"

use "C:\Colin Westminster\EST Field Trials\STATA Heat Pump Analysis\ashp", clear

table ID, contents(sum v17) // display table showing tot Space Heating circulation pump input

by ID: egen SpaceHtgPump = sum(v17)

//////////////////////////
//  STATA 12

//  09 November 2012

//  SET UP ashp_bysyst_metered for SPF full analysis.
//  Change all VARIABLE NAMES to match schematics in "EST Heat Pump Trial Site Report" to column headings v1, v2, v3, etc.
//  Set up VARIABLE LABELS to match EXCEL column headers

//  Use "LINN TAX for Macro Paper" SPF analysis

//  Rename and label all variables to simplify equations

use "/Users/gleesoc/Google Drive/HEAT PUMP RESEARCH/EST Field Trials/STATA 12_11_09/ashp_bysyst_metered.dta"

. rename COP SPFH2

. rename SPF SPFH5

. replace SPFH2 = . if SPFH2 == 0

 label var HWFlowTot "DHW Flow Total (litres)"

. label var SHEnergyTot "Space Heating Output Total (Wh)"

. label var Immersion "Immersion heater Total (Wh)"

. label var WhBoost "External Boost Total (Wh)"

. label var HP2EnergyIn "2nd ASHP Energy In Total (Wh)"

. label var ASHPEnergyOut "ASHP Energy Out (Wh)"

. . label var ASHPEnergyOut "ASHP Energy Out Total (Wh)"

. label var ASHPEnergyIn "ASHP Energy In Total (Wh)

. label var DHWEnergyOut "DHW Energy Out Total (Wh)

. label var PrimaryEnergy "Primary Flow Energy Total (Wh)"

. label var SPFH2 "SPFH2"

. label var SPFH5 "SPFH5"

. . drop  remove

. rename HWFlowTot v20

. rename SHEnergyTot v21

. rename ASHPEnergyOut v24

. rename Immersion v18

. rename WhBoost v26

. rename HP2EnergyIn v27

. rename ASHPEnergyIn v16

. rename DHWEnergyOut v20
v20 already defined
r(110);

. rename HWFlowTot v19
```

```
variable HWFlowTot not found
r(111);

. rename v20 v19

. rename DHWEnergyOut v20

. rename PrimaryEnergy v28

//  GENERATE NEW VARIABLES BASED ON MOST COMMON EQUATION

generate SPFH2 = v24/v16

replace SPFH2 = v24/v16

. replace SPFH2 = v24/v16
(15 real changes made)

end of do-file

. rename SPFH5 SPFhps

. replace SPFhps = v24/v16
(19 real changes made)

. label var SPFhps "SPFhps"

. generate SPFH4 = v24/(v16+v26)

. generate SPFH5 = (v20+v21)/(v16+v18)

//  NOW EDIT SPF EQUATIONS by, for example, replace SPFH2 = (v24/v16) if ID == 400 | ID == 401

replace SPFhps = v21/v16 if ID == 429 | ID == 478 | ID == 479

replace SPFH4 = v21/(v16+v26) if ID == 429 | ID == 479

replace SPFH4 = v24/v16 if ID == 472

replace SPFH4 = v24/(v16+v18) if ID == 473

replace SPFH4 = v21/(v16+v18) if ID == 478

replace SPFH5 = (v20+v21)/(v16+v18+v26) if ID == 424 | ID == 486 | ID == 487

replace SPFH5 = (v20+v21+v28)/v16 if ID == 425

replace SPFH5 = (v20+v21)/v16 if ID == 440 | ID == 488

replace SPFH5 = (v20+v21)/(v16+v27) if ID == 441

//  THE FILE IS NOW READY TO REMOVE INAPPROPRIATE ENTRIES DUE TO METERING AND BOOST

replace SPFH2 = . if ID == 424 | ID == 425 | ID == 429 | ID == 440 | ID == 441 | ID == 442 | ID == 443 | ID == 444 | ID == 445 | ID == 446 | ID == 447 | ID == 472 | ID == 473 | ID == 474 | ID == 475 |ID == 478 | ID == 477 | ID == 479 | ID == 486 | ID == 487 | ID == 488

replace SPFhps = . if ID == 418 | ID == 422 | ID == 423 | ID == 424 | ID == 425 | ID == 426 | ID == 440 | ID == 441 | ID == 444 | ID == 472 | ID == 473 | ID == 475 | ID == 486 | ID == 487 | ID == 488

replace SPFhps = 1.79 if ID == 477

replace SPFH4 = . if ID == 418 | ID == 422 | ID == 423 | ID == 425 | ID == 426 | ID == 440 | ID == 441 | ID == 442 | ID == 443 | ID == 444 | ID == 445 | ID == 446 | ID == 447 | ID == 473 ID == 477 | ID == 486 | ID == 487 | ID == 488

replace SPFH5 = . if ID == 429 | ID == 442 | ID == 443 | ID == 444 | ID == 445 | ID == 446 | ID == 447 | ID == 472 | ID == 474 | ID == 475 | ID == 477 | ID == 478 | ID == 479

drop ID == 483

drop if ID == 483

replace SPFhps = v22/v16

table ID, contents(sum v22) // display table showing output from 477
```

```
by ID: egen v22 = sum(v22)

table ID, contents(sum v22) // display table showing tot Primary F&R Energy

drop if ID == 489 | ID == 490

replace SPFH4 = . if ID == 477

replace SPFH5 = . if ID == 477

////////////////////////////////////////////////////////////////////////////////

// 28 November 2012  Insert SPFH3 in ASHP by using the mean circulation pump energy found in v17 GSHP of 166426 Wh/year

//  generate SPFH3 = v24/(v16 - 166426)

generate SPFH3 = v24/(v16-166426)

replace SPFH3 = . if ID == 425 | ID == 440 | ID == 441 | ID == 472 | ID == 486 | ID == 487 | ID == 488

replace SPFH3 = . if ID | 477

replace SPFH3 = v21/((v16-166426)+v18) if ID == 478

replace SPFH3 = v21/((v16-166426)+v26) if ID == 479

replace SPFH3 = v24/(v16-166426)

replace SPFH3 = . if ID == 425 | ID == 440 | ID == 441 | ID == 472 | ID == 486 | ID == 487 | ID == 488

replace SPFH3 = . if ID | 477

replace SPFH3 = v21/((v16-166426)+v18) if ID == 478

replace SPFH3 = v21/((v16-166426)+v26) if ID == 479

save "/Users/gleesoc/Google Drive/HEAT PUMP RESEARCH/EST Field Trials/STATA 12_11_09/ashp with SPFH3.dta" replace

replace SPFH3 = v24/(v16-166426)

replace SPFH3 = . if ID | 477

replace SPFH3 = v21/((v16-166426)+v18) if ID == 478

replace SPFH3 = v21/((v16-166426)+v26) if ID == 479

replace SPFH3 = v24/(v16-166426)

replace SPFH3 = v21/((v16-166426)+v18) if ID == 478

replace SPFH3 = v21/((v16-166426)+v26) if ID == 479

replace SPFH3 = v24/(v16-166426) if ID == 445

replace SPFH3 = v24/((v16-166426)+v27) if ID == 441

replace SPFH3 = v24/(v16-166426) if ID == 472

replace SPFH3 = v24/((v16-166426)+v18) if ID == 426

/////////////////////////////////////////////////////////////////////

///  Generate SEEF to match EST Trial outputs

/// Tuesday 23 April 2013

//  STATA 12

generate SEEF = SPFH5

//  NOW EDIT SPF EQUATIONS by, for example, replace SPFH2 = (v24/v16) if ID == 400 | ID == 401

replace SEEF = v21/(v16+v18) if ID == 478 | ID == 479
```

```
replace SEEF = v24/v16 if ID == 442 | ID == 443 | ID == 444 | ID == 445 | ID == 446 | ID == 447 | ID == 472

replace SEEF = v24/(v16+v26) if ID == 474 | ID == 475

replace SEEF = v21/(v16+v26) if ID == 429

///////////////////////////////////////////////////////////////////////////////////////

//  STATA 12

/// 1st May 2013
//  Corrections:

////////////////////////////////////////////////////////////////////////////////////

//   RECOGNISE THAT SPFH4 IS DIFFERENT FROM SPFH5

//  SPFH4 is Qout/Qin whereas SPFH5 is (Qout - Cylinder losses)/Qin

////////////////////////////////////////////////////////////////////////////////////

by ID: egen v24 = sum(v24)

replace SPFH2 = v24/v16 if ID == 418 | ID == 422 | ID == 423 | ID == 426

replace SPFhps = .

replace SPFhps = v24/v16 if ID == 440 | ID == 441 | ID == 442 | ID == 443 | ID == 444 | ID == 445 | ID == 446 | ID == 473

replace SPFH4 = v24/(v16+v26) if ID == 424 | ID == 429 | ID == 447 | ID == 474 | ID == 475

replace SPFH4 = v24/v16 if ID == 472 | ID == 473

replace SPFH4 = v21/v16 if ID == 478 | ID == 479

drop SPFH3

drop if ID == 477

replace SPFH5 = . if ID == 447

replace SPFH4 = v21/(v16+v26) if ID == 429

replace SEEF = v24/(v16+v26) if ID == 447

/////////////////////////////////////////////////////////////////////////////////////

/// 14 June 2013

/// Adjust SPFH4 due to v18 being both 100% input and output, eg, 21+28+18/16+18

////////////////////////////////////////////////////////////////////////////////////

replace SPFH4 = (v24+v18)/(v16+v18) if ID == 423 | ID == 424 | ID == 426

replace SPFH4 = v21/(v16+v26) if ID == 429 | ID == 479

replace SPFH4 = v24/v16 if ID == 442 | ID == 443 | ID == 444 | ID == 445 | ID == 446 | ID == 472

replace SPFH4 = (v24+v26)/(v16+v26) if ID == 447

replace SPFH4 = (v24+v28)/(v16+v18) if ID == 473

replace SPFH4 = v24/(v16+v26) if ID == 474 | ID == 475

replace SPFH4 = v21/(v16+v18) if ID == 478

replace SPFhps = v24/(v16+v27) if ID == 441

replace SPFH5 = (v20+v21)/(v16+v18) if ID == 423
```

STATA example GSHP data analysis ‘Do-file”

```
. use "/Users/gleesoc/Google Drive/HEAT PUMP RESEARCH/EST Field Trials/STATA Analysis 2012/gshp.dta", clear

sort ID time

// HW Flow Tot Time

//table ID, contents(sum v19) // display table showing tot HW Flow
cap drop HWFlowTot
by ID: egen HWFlowTot = sum(v19)

//NEW ENTRIES

// SH Energy Tot Time

//table ID, contents(sum v21) // display table showing tot SH Energy
cap drop SHEnergyTot
by ID: egen SHEnergyTot = sum(v21)

// GSHP Energy Out

//table ID, contents(sum v22) // display table showing tot GSHP Total Output
cap drop GSHPEnergyOut
by ID: egen GSHPEnergyOut = sum(v22)

// Electrical input/Immersion

//table ID, contents(sum v18) // display table showing tot Electrical input/Immersion
cap drop Immersion
by ID: egen Immersion = sum(v18)

// 2nd SH system Energy Tot

//table ID, contents(sum v29) // display table showing tot 2nd SH Energy
cap drop SecSHEnergyTot
by ID: egen SecSHEnergyTot = sum(v29)

// Heat Pump Tot Power Input

//table ID, contents(sum v16) // display table showing tot GSHP Tot Input
cap drop GSHPEnergyIn
by ID: egen GSHPEnergyIn = sum(v16)

// Hot Water Energy

//table ID, contents(sum v20) // display table showing tot DHW Tot Output
cap drop DHWEnergyOut
by ID: egen DHWEnergyOut = sum(v20)

// Primaries energy

//table ID, contents(sum v28) // display table showing tot Primary F&R Energy
cap drop PrimaryEnergy
by ID: egen PrimaryEnergy = sum(v28)

// Wh Boost

//table ID, contents(sum v26) // display table showing tot Wh Boost
cap drop BoostEnergy
by ID: egen BoostEnergy = sum(v26)

//////////////////////////////////////////////////////////////////////////////////////////

// SpaceHtgPump

//table ID, contents(sum v17) // display table showing tot SpaceHtgPump

cap drop SpaceHtgPump
by ID: egen SpaceHtgPump = sum(v17)

// generating efficiency scores
cap drop SPF
```

```
// (20+21)/(16+18)
gen SPF = (DHWEnergyOut + SHEnergyTot) / (GSHPEnergyIn + Immersion)
// fix for any anomalies

// (20+21+29)/((16+18)
replace SPF = (DHWEnergyOut + SHEnergyTot + SecSHEnergyTot) / (GSHPEnergyIn + Immersion) if ID == 469 // Change formula
for the system to include v29 in efficiency, this is an anomoly for this ID only [27 = 29 in excel file]

// (20+21)/(16+18+26)
replace SPF = (DHWEnergyOut + SHEnergyTot) / (GSHPEnergyIn + Immersion + BoostEnergy) if ID == 481 | ID == 482

// (20+21)/16
replace SPF = (DHWEnergyOut + SHEnergyTot) / GSHPEnergyIn if ID == 413 | ID == 415 | ID == 416 | ID == 417 | ID == 419 | ID ==
420 | ID == 421 | ID == 432 | ID == 433 | ID == 434 | ID == 435 | ID == 436 | ID == 437 | ID == 438 | ID == 439 | ID == 453 | ID == 454
| ID == 455 | ID == 456 | ID == 458 | ID == 459 | ID == 466 | ID == 471 | ID == 491

// 21/16
replace SPF = SHEnergyTot / GSHPEnergyIn if ID == 430 | ID == 431 | ID == 492 // Change formula for system due to Independent
DHW

// 22/16
replace SPF = GSHPEnergyOut / GSHPEnergyIn if ID == 457 | ID == 470 // Change due to use of heat meter 22

// 16/22 System 477 where ASHP produces virtually nothing. Treat as GSHP only

replace SPF = GSHPEnergyOut / GSHPEnergyIn if ID == 477

// 414 has swimming pool on v28

replace SPF = (DHWEnergyOut + SHEnergyTot + PrimaryEnergy) / GSHPEnergyIn if ID == 414

// Need to add system 477 and combine (427 + 428) and (450 + 468)

cap drop COP
gen COP = GSHPEnergyOut / GSHPEnergyIn
// fix for any anomalies

replace COP = SHEnergyTot / GSHPEnergyIn if ID == 430 | ID == 431

replace COP = (SHEnergyTot + PrimaryEnergy) / GSHPEnergyIn if ID == 460 | ID == 461 | ID == 462 | ID == 463 | ID == 464 | ID ==
465 | ID == 471

// checking what we have generated
//table ID, contents(min SPF min COP)

// creating smaller file with info on each system only no time data
// remove preserve
drop v* time
by ID: keep if _n == 1

// add an extra row at the bottom of dataset
local n = _N + 1
set obs `n'
// set ID number to be 927 in this last row
replace ID = 927 in `n'
// set HWFlowTot in the last row to be sum of values in 15th and 16th rows
replace HWFlowTot = HWFlowTot[15] + HWFlowTot[16] in `n'

// Add values for (427 + 428) = 927

replace SHEnergyTot = SHEnergyTot[15] + SHEnergyTot[16] in `n'

replace DHWEnergyOut = DHWEnergyOut[15] + DHWEnergyOut[16] in `n'

replace GSHPEnergyOut = GSHPEnergyOut[15] + GSHPEnergyOut[16] in `n'

replace Immersion = Immersion[15] + Immersion[16] in `n'

replace SecSHEnergyTot = SecSHEnergyTot[15] + SecSHEnergyTot[16] in `n'

replace GSHPEnergyIn = GSHPEnergyIn[15] + GSHPEnergyIn[16] in `n'

replace PrimaryEnergy = PrimaryEnergy[15] + PrimaryEnergy[16] in `n'
```

```
replace BoostEnergy =   BoostEnergy[15] + BoostEnergy[16] in `n'

//Generate SPF and COP
replace SPF = (DHWEnergyOut + SHEnergyTot) / (GSHPEnergyIn + Immersion) if ID == 927

replace COP = GSHPEnergyOut / GSHPEnergyIn if ID == 927

// add an extra row at the bottom of dataset
local n = _N + 1
set obs `n'
// set ID number to be 950 in this last row
replace ID = 950 in `n'
// set HWFlowTot in the last row to be sum of values in 27th and 45th rows
replace  HWFlowTot =  HWFlowTot[27] +  HWFlowTot[45] in `n'

// Add values for (450 + 468) = 950

replace  SHEnergyTot =  SHEnergyTot[27] + SHEnergyTot[45] in `n'

replace DHWEnergyOut =   DHWEnergyOut[27] + DHWEnergyOut[45] in `n'

replace  GSHPEnergyOut =  GSHPEnergyOut[27] + GSHPEnergyOut[45] in `n'

replace  Immersion =   Immersion[27] + Immersion[45] in `n'

replace  SecSHEnergyTot =   SecSHEnergyTot[27] +  SecSHEnergyTot[45] in `n'

replace  GSHPEnergyIn =    GSHPEnergyIn[27] +  GSHPEnergyIn[45] in `n'

replace  PrimaryEnergy =   PrimaryEnergy[27] +  PrimaryEnergy[45] in `n'

replace BoostEnergy =   BoostEnergy[27] + BoostEnergy[45] in `n'

//Generate SPF and COP

replace SPF = (DHWEnergyOut + SHEnergyTot) / (GSHPEnergyIn + Immersion + BoostEnergy) if ID == 950

replace COP = GSHPEnergyOut / GSHPEnergyIn if ID == 950

//Remove non-relevant files 427, 428, 450, 468

// Removing no-relevant files.  Note these are not removed but marked as ZERO (0)
gen remove = 1

replace remove = 0 if ID == 427 |  ID == 428 |  ID == 450 |  ID == 468

//////////////////////////////////////////////////////////////////////////////////////////////

save "/Users/gleesoc/Google Drive/HEAT PUMP RESEARCH/EST Field Trials/STATA Analysis 2012/gshp_byid.dta", replace
restore
/////////////////////////////////////////////////////////////////////////////////////////////////

// SET COP AND SPF TO ZERO OR NEW FORMULA DUE TO SYSTEM SET UP

replace COP = 0 if ID == 427 | ID == 428 | ID == 430 | ID == 462 | ID == 463 | ID == 465 | ID == 468

replace COP = (SHEnergyTot + PrimaryEnergy) / GSHPEnergyIn if ID == 491

replace COP = SHEnergyTot / GSHPEnergyIn if ID == 470

replace SPF = 0 if ID == 427 | ID == 428 | ID == 450 | ID == 468

replace SPF = (DHWEnergyOut + SHEnergyTot) / (GSHPEnergyIn + Immersion) if ID == 456

replace SPF = SHEnergyTot / GSHPEnergyIn if ID == 470

replace SPF = GSHPEnergyOut / GSHPEnergyIn if ID == 477

////////////////////////////////////////////////////////////////////////////////////////////
save

// also SAVED IN  "/Users/gleesoc/Google Drive/HEAT PUMP RESEARCH/EST Field Trials/STATA 12_11_09/GSHP/gshp_byid.dta"
```

```
// STATA 12

// 25 November 2012

// SET UP gshp_byid for SPF full analysis.
// Change all VARIABLE NAMES to match schematics in "EST Heat Pump Trial Site Report" to column headings v1, v2, v3, etc.
// Set up VARIABLE LABELS to match EXCEL column headers

// Use "LINN TAX for Macro Paper" SPF analysis

. use "/Users/gleesoc/Google Drive/HEAT PUMP RESEARCH/EST Field Trials/STATA 12_11_09/GSHP/gshp_byid.dta"

// Rename and label all variables to simplify equations

. rename SPF SPFH2

. rename COP SPFhps

. replace SPFH2 = . if SPFH2 == 0

 label var HWFlowTot "DHW Flow Total (litres)"

. label var SHEnergyTot "Space Heating Output Total (Wh)"

. label var GSHPEnergyOut "GSHP Energy Out Total (Wh)"

. label var Immersion "Immersion heater Total (Wh)"

label var SecSHEnergyTot "2nd Space Heating Total (Wh)"

. label var GSHPEnergyIn "GSHP Energy In Total (Wh)

. label var DHWEnergyOut "DHW Energy Out Total (Wh)"

. label var PrimaryEnergy "Primary Flow Energy Total (Wh)"

. label var BoostEnergy "External Boost Total (Wh)"

. label var SpaceHtgPump "Space Heating Circ Pump"

. label var SPFH2 "SPFH2"

. label var SPFH5 "SPFH5"

drop  remove

. rename HWFlowTot v19

. rename SHEnergyTot v21

. rename GSHPEnergyOut v24

. rename Immersion v18

. rename SecSHEnergyTot v29

. rename GSHPEnergyIn v16

. rename DHWEnergyOut v20

. rename PrimaryEnergy v28

. rename BoostEnergy v26

. rename SpaceHtgPump v17

// GENERATE NEW VARIABLES BASED ON MOST COMMON EQUATION

replace SPFH2 = v24/v16
replace SPFhps = v24/v16

. generate SPFH4 = v24/(v16+v26)

. generate SPFH5 = (v20+v21)/(v16+v18)
```

// NOW EDIT SPF EQUATIONS by, for example, replace SPFH2 = (v24/v16) if ID == 400 | ID == 401

replace SPFH2 = (v21+v28)/v16 if ID == 460 | ID == 461 | ID == 462 | ID == 463 | ID == 464 | ID == 465

replace SPFH2 = v21/v16 if ID == 469 | ID == 470

replace SPFH2 = . if ID == 430 | ID == 431 | ID == 432 | ID == 433 | ID == 434 | ID == 435 | ID == 436 | ID == 437 | ID == 438 | ID == 43

replace SPFH2 = . if ID == 450 | ID == 451 | ID == 452 | ID == 453 | ID == 454 | ID == 455 | ID == 456 | ID == 458 | ID == 459

replace SPFH2 = . if ID == 466

replace SPFH2 = . if ID == 467 | ID == 468 | ID == 471 | ID == 476 | ID == 477

replace SPFH2 = . if ID == 480 | ID == 481 | ID == 482 | ID == 491 | ID == 492 | ID == 927 | ID == 950

replace SPFH2 = . if ID == 439 | ID == 407 | ID == 408 | ID == 409 | ID == 410 | ID == 411 | ID == 412 | ID == 413 | ID == 414 | ID == 415

replace SPFH2 = . if ID == 416 | ID == 417 | ID == 419 | ID == 420 | ID == 421 | ID == 427 | ID == 428

//NOTE there are no entries for v22 457 exists at SPFH2//

//replace SPFH2 = v22/v16 if ID == 409 | ID == 411 with data entries directly into data editor

//Edit SPFhps

replace SPFhps = .

replace SPFhps = v21/v16 if ID == 430

replace SPFhps = (v21+v28)/(v16+v17) if ID == 460 | ID == 461 | ID == 462 | ID == 463 | ID == 464 | ID == 465

replace SPFhps = (v21+v28+v29)/(v16+v17) if ID == 469

replace SPFhps = v21/(v16+v17) if ID == 470

// 471 from Worcester has max v16 of 1032Wh = 12 kW. Cannot be SPFH2 or SPFhps

//NOTE there are no entries for v22

//replace SPFhps = v22/(v16+v17) if ID == 457 with data entries directly into data editor

//Edit SPFH4
replace SPFH4 = .

replace SPFH4 = (v21+v28)/(v16+v17) if ID == 460 | ID == 461 | ID == 462 | ID == 463 | ID == 464 | ID == 465 | ID == 491

replace SPFH4 = v21/(v16+v17+v26) if ID == 431

replace SPFH4 = (v21+v28)/v16 if ID == 451 | ID == 452 | ID == 471

replace SPFH4 = v21/v16 if ID == 453 | ID == 492 | ID == 430

replace SPFH4 = (v21+v28+v29+v18)/(v16+v17+v18) if ID == 469

replace SPFH4 = v21/(v16+v17) if ID == 470

//NOTE there are no entries for v22
// replace SPFH4 = v22/(v16+v17) if ID == 457

//Edit SPFH5

replace SPFH5 = (v20+v21)/v16

replace SPFH5 = (v20+v21)/(v16+v18) if ID == 407 | ID == 408 | ID == 409 | ID == 410 | ID == 411 | ID == 412 | ID == 451 | ID == 452 | ID == 456 | ID == 467 | ID == 476

replace SPFH5 = (v20+v21+v28)/(v16+v17) if ID == 414

```
replace SPFH5 = (v20+v21)/(v16+v17+v18) if ID == 460 | ID == 461 | ID == 462 | ID == 463 | ID == 464 | ID == 465

replace SPFH5 = (v20+v21+v29)/(v16+v17+v18) if ID == 469

replace SPFH5 = (v20+v21)/(v16+v18+v26) if ID == 481 | ID == 482

replace SPFH5 = (v20+v21)/(v16+v17) if ID == 491

replace SPFH5 = (v20+v21)/(v16+v17+v26) if ID == 950

replace SPFH5 = . if ID == 430 | ID == 431 | ID == 457 | ID == 470 | ID == 477 | ID == 492

replace SPFhps = (v21+v28)/v16 if ID == 451 | ID == 452

replace SPFhps = v21/v16 if ID == 467

replace SPFH4 = v21/v16 if ID == 467

replace SPFhps = v24/v16 if ID == 477
replace SPFhps = v21/v16 if ID == 492

replace SPFH2 = (v21+v28+v29)/v16 if ID == 469

replace SPFhps = (v21+v28+v29)/(v16+v17) if ID == 469

replace SPFH4 = (v21+v28+v29)/(v16+v17+v18) if ID == 469

replace SPFH5 = (v20+v21+v29)/(v16+v17+v18) if ID == 469

replace SPFhps = . if ID == 467

replace SPFhps = (v21+v28)/v16 if ID == 467

//  467 gives spf of 0.5 for hps, H4 and H5.  Remove it.

replace SPFhps = . if ID == 467

replace SPFH4 = . if ID == 467

replace SPFH5 = . if ID == 467

//  450 should be removed since itis combined with 468 and shown as 950

replace SPFH5 = . if ID == 450

//  950 only gives SPFH5

replace SPFH5 = (v20+v21)/(v16+v18) if ID == 950

replace SPFH2 = . if ID == 950

replace SPFH5 = (v20+v21)/(v16+v18) if ID == 927

//  927 is extremely high spf.  Value for SPFH2 is 5.59.  3 months of missing data.

replace SPFH2 = 24667300/4409205 if ID == 927

replace SPFhps = 24667300/(4409205+214539+279687) if ID == 927

replace SPFH4 = 24667300/(4409205+214539+279687+254705) if ID == 927

replace SPFH5 = (1910200+22098800)/(4409205+214539+279687+254705) if ID == 927

replace SPFH2 = (v28+v21+v29)/v16 if ID == 469

replace SPFhps = (v21+v28+v29)/(v16+v17) if ID == 469

replace SPFH4 = (v21+v28+v29)/(v16+v17+v18) if ID == 469

replace SPFH5 = (v20+v21+v29)/(v16+v17+v18) if ID == 469

//  Generate SPFH3 from values for v17

replace v17 = . if v17 == 0
```

```
replace v17 = . if ID == 491

// mean v17 = 166426.4 Wh or 166 kWh/year

//  median v17 =  134272 or 134 kWh/yr

//  use mean to calculate SPFH3 in ASHP


//  STATA 12
/// 1st May 2013
//  Corrections:

////////////////////////////////////////////////////////////////////////////////

//   RECOGNISE THAT SPFH4 IS DIFFERENT FROM SPFH5

//  SPFH4 is Qout/Qin whereas SPFH5 is (Qout - Cylinder losses)/Qin

////////////////////////////////////////////////////////////////////////////////

replace SPFH4 = . if ID == 453

replace SPFhps = . if ID == 451 | ID == 452

replace SPFH4 = (v21+v28+v29)/(v16+v17) if ID == 469

replace SPFH5 = (v20+v21+v29)/(v16+v17+v18) if ID == 469

replace SPFH4 = 19096800/(v16+v17) if ID == 457

replace SPFH4 = v21/(v16+v17) if ID == 470

replace SPFH5 = (v20+v21)/(v16+v18) if ID == 467
// note that 467 has an SPF of SPFH5 = 0.57 therefore DROP

drop if ID == 467

replace SPFH4 = SPFhps if ID == 477

// Get rid of separate input rows for double systems

drop if ID == 427 | ID == 428

drop if ID == 450 | ID == 468

///  Generate SEEF to match EST Trial outputs

generate SEEF = SPFH5

//  NOW EDIT SPF EQUATIONS by, for example, replace SEEF = v21/(v16+v18) if ID == 478 | ID == 479

replace SEEF = SPFH4 if ID == 430 | ID == 431 | ID == 457 | ID == 477 | ID == 492

//////   Adjust for SPFH4 based on immersion input = immersion output 14 June 2013

replace SPFH2 = . if ID == 409 | ID == 411

replace SPFH4 = (v22+v18)/(v16+v18) if ID == 409 | ID == 411

replace SPFH4 = v21/v16 if ID == 430 | ID == 453 ID == 492

replace SPFH4 = v21/(v16+v17+v26) if ID == 431

replace SPFH4 = (v21+v28+v18)/(v16+v18) if ID == 451

replace SPFH4 = (v21+v28)/v16 if ID == 452 | ID == 471

replace SPFH4 = v22/(v16+v17) if ID == 457

replace SPFH4 = (v21+v28+v18)/(v16+v17+v18) if ID == 460 | ID == 461 | ID == 462 | ID == 463 | ID == 464 | ID == 465

replace SPFH4 = (v21+v28+v29)/(v16+v17+v18) if ID == 469
```

```
replace SPFH4 = v21/(v16+v17) if ID == 470

replace SPFH4 = (v21+v28)/(v16+v17) if ID == 491


/// PLAYING
//Kensa 457

. tsset time

// compare v16 to v1 outdoor temp

graph twoway (tsline v16 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(1)) (tsline v1 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(2))

/// Kensa 470

. tsset time

//////  compare v16 to v1 outdoor temp

graph twoway (tsline v16 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(1)) (tsline v1 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(2))

/// Kensa 430

. tsset time

// compare v16 to v1 outdoor temp

graph twoway (tsline v16 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(1)) (tsline v1 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(2))

//////
/////  v16 versus return temperature

//Kensa 457

. tsset time

// compare v16 to v3 lounge temp

graph twoway (tsline v16 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(1)) (tsline v12 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(2))

/// Kensa 470

. tsset time

// compare v16 to v3 lounge temp

graph twoway (tsline v16 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(1)) (tsline v12 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(2))

/// Kensa 430

. tsset time

// compare v16 to v3 lounge temp

graph twoway (tsline v16 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(1)) (tsline v12 if tin(06mar201000:00:00, 06mar201023:59:00), yaxis(2))

/// 2 day run time

tsset

graph twoway (tsline v16 if tin(06mar201000:00:00, 07mar201023:59:59), yaxis(1)) (tsline v1 v3 if tin(06mar201000:00:00, 07mar201023:59:59), yaxis(2))
```